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PRODUCT ENGINEERING

JANUARY
1935

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Time turns the page and a new calendar year begins. But a sound policy of developing new products stops neither for dates nor days, nor for seasons of the year. It is continuous so that its accomplishments may be at hand when the market is ready

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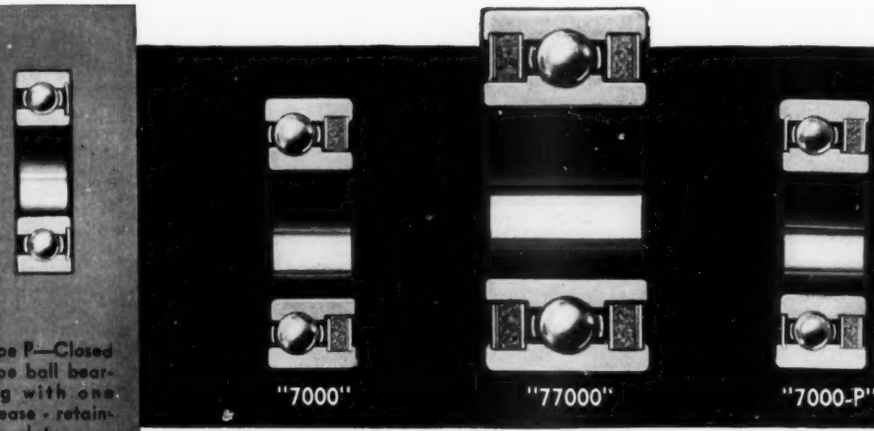
Mechanical Design • Product Development

McGRAW-HILL PUBLISHING COMPANY, INC.



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Out of their experience of more than 23 years, NORMA-HOFFMANN engineers — pioneers in the design of enclosed and self-protected bearings —have developed and patented a range of types which meets practically every bearing condition and which affords the mechanical world

The Most Complete Line of SELF-PROTECTED BEARINGS in America

The "GREASEAL" Series of Felt-Protected Ball Bearings—in the three types illustrated above—is marked by the following outstanding features which make for better performance and more lasting satisfaction:- thick, closely-fitting felts between removable plates forming an effective labyrinth against the recessed inner ring - - - FELT SEAL REMOVABLE in its entirety for inspection, cleaning or renewal of grease - - - wide, solid inner and outer rings, with maximum contact on shaft and housing, make inserts in housing unnecessary and militate against slippage, looseness, and escape of lubricant past outer ring - - - felt seal within confines of both rings and not exposed to injury - - - seal construction avoids race distortion, assuring dimensional exactness and quiet running - - - grease capacity ample for long periods of service. . . . . Eight other types of Self-Protected PRECISION Bearings are here pictured and indexed. Write for the complete Catalog. Let our engineers aid you in selection and application.

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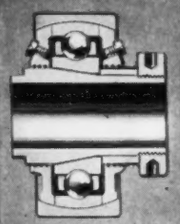
Type P—Closed type ball bearing with one grease - retaining plate.

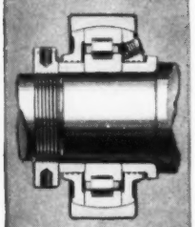
Type PP—closed type ball bearing with two grease - retaining plates.

Type GS—"Greaseal" felt - protected, grease - packed ball bearing.

Type EGS—Adjustable, felt - protected, grease - packed ball bearing.

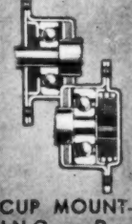


Type LTW—Self - aligning ball bearing with swiveling grease - retaining side plates and adapter sleeve.



Type RLSW—Self - aligning roller bearing with swiveling grease - retaining side plates.

Type RLTW—Self - aligning roller bearing with swiveling grease - retaining side plates and adapter sleeve.



CUP MOUNTING—Designed to use with small open type ball bearings in high-speed devices; furnished as complete units.

"NORMA-HOFFMANN"

PRECISION BEARINGS

BALL, ROLLER AND THRUST

PRODUCT ENGINEERING

VOLUME 6 NUMBER 1

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Next Month

Automotive engineers are generally credited with being the leaders in the development of products of merit and appeal. In the next number will be analyzed the successive steps in automobile improvements over the past five years with reference to the material used and types of construction. By charts, tables and pictures will be shown the results obtained through the adoption of various metals and alloys, the part played by welding, new types of finishes and the extended use of die-castings, rubber, molded plastics and other materials.

In February we also give the first of a series of articles that show how to reduce vibration problems to the simplest terms and calculate the critical speeds without recourse to complicated mathematics.

In this number is an article that emphasizes the value of well-designed built-in lighting for machines. How to design for such lighting will be the subject of an article appearing in the next number.

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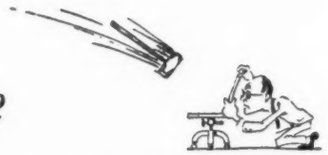
Intimate Correspondence



Percentages

PRODUCT ENGINEERING
NEW YORK

Relative to your Survey of Materials in the November number of *Product Engineering*—there seems to be something lacking in the tables. For instance, in the table "Why alloy cast irons are being used," as on page 404—how was the item, "Total for all companies," arrived at?

In other tables, for example, 2 per cent of the number of companies written to reported hard facing, and 11 per cent bronze. What did the other 87 per cent use? Also how about percentages as 57, 41 and 61, giving a sum of 159 per cent?

It is believed that the total number of companies written to should be given, and the number of replies, from which could be figured out the percentages.

CHAS. H. HUGHES,
New Dorp, N. Y.

THESE are fair questions. Here is the way we went at these particular calculations: In the table, "Why alloy cast irons are being used," page 404, the item "Total for all companies," was found by taking all of the companies that sent in answers and figuring out what percentage of them were using alloy cast irons for the reasons mentioned. Other percentages were figured the same way.

Where percentages do not add up to 100, the fact is that not all of the companies in a particular classification gave answers. Also, some companies reported use of more than one material.

These are just explanations, not excuses, and we are planning now to set up our ngures in the next survey so that there can be no question of the interpretation.

Indexed

PRODUCT ENGINEERING
NEW YORK

We would like to secure copies of the articles listed in the Index to Volume 5 of *Product Engineering*, under the title "Plastics," as follows:

"Designing with laminated plastics," H. M. Richardson ..... p. 18
"Engineering data for designing molded plastics," F. W. Warner, pt. 1 p. 138; pt. 2 ..... p. 173
"Laminated plastic sheets with printing" ..... p. 429
"Plaskon parts for hardware" ..... p. 428

Your cooperation in assisting us to secure the above will be greatly appreciated.

C. A. PRICE
New York City

AND that, of course, is what an index is for. You look it over, pick out what you missed during the year, and then arrange to complete your file.

Cantilevers

PRODUCT ENGINEERING
NEW YORK

I should appreciate your sending me, if possible, two reprints of "Cantilever Spring Design" by R. L. Peek, Jr., appearing on page 439 of *Product Engineering* for November, 1934.

W. B. HUSTON
West Orange, N. J.

IT IS surprising the number of readers who are asking for reprints and clippings of spring articles and reference sheets. Ought to mean good business for the fellows who make springs or sell spring stock—particularly if they advertise in *Product Engineering*.

No Charge

PRODUCT ENGINEERING
NEW YORK

Will you kindly forward marked for the writer's attention a clipping of the article entitled "Safe Loads for Journal Bearing" by Victor Tartarinoff which appears in the December issue of *Product Engineering*.

We understand there should be no charge for the clipping involved. If we are mistaken, kindly advise.

W. H. RYAN
Neenah, Wis.

BEING a good purchasing agent Mr. Ryan naturally checks up on prices first. But this time it wasn't necessary for there is no charge for clippings—as long as they last.

Overlaps

PRODUCT ENGINEERING
NEW YORK

I have been a subscriber to *Product Engineering* since it started and, like some other readers, index and file all good articles and sets of data. There has been a great deal of improvement in arrangement of articles that contain valuable data so that in filing the pages one subject does not interlock with an article of an entirely different subject causing confusion in filing. However, in the October, 1934 issue there is an overlapping and I would like for you to send me the full two pages 382 and 383. In my filing I cut everything 8½x11 in. and perforate to fit standard 3-ring filing books so I like the size of the magazine page.

May I suggest that at some time in the future you print a book or folder combining all such data from all back numbers like that shown on pages 390 and 391 of the October, 1934 issue?

Thanking you for these helps, I remain,

HARRY R. DOWSON
Reiffton, Pa.

AS REMARKED before on this page we are trying hard to keep clippable material separate but once in a while we slip, as in this case. If you speak soon enough clippings from our files will take care of you.

Modernize

PRODUCT ENGINEERING
NEW YORK

I am sending in a renewal of my subscription for *Product Engineering* for another year.

Of all the technical and industrial publications which pass over my desk I am convinced *Product Engineering* is, in many respects, the most practical, interesting and modern. I believe you are effectively sowing the seeds for renewed industrial activity. We find nowadays a great many people, some of them in high positions, deploring the fact that at this particular time there is no great new industry forthcoming to raise industry out of its despondency as has happened in the past. Personally, I believe it is just as well that there is not a brand new industry just around the corner for the reason that mature industry now has an opportunity to modernize itself in accordance with new basic materials.

When we consider the new steel alloys, the nickel, cobalt, tungsten and chromium alloys, and the new aluminum alloys which are light, strong, tough and non-corrosive, and the die casting alloys; the new plastic molding materials which in the finished product present beautiful glowing finishes and strong, light, durable bodies, acid proof and non-inflammable, we are brought to the realization that a large percentage of our old machinery and products has become antiquated. There are so many advantages to be gained in savings in weight, space, efficiency, speed, freedom from noise and vibration, and also in appearance by employing the modern basic materials that it appears that it would pay to redesign practically all of our old machinery on new lines.

To be sure, the N.R.A. discourages new machinery, but what can be expected from politicians and Doctors of Sociology? In the end, the ball will be passed to the engineers again, as it always has been in the past, and the necessary drive toward the goal will be well under way.

B. D. WILLIS
Chicago, Ill.

WE ALMOST used this letter for an editorial but decided that it would be fairer to let Mr. Willis tell his own story. He's absolutely right, particularly in that last sentence.

PRODUCT ENGINEERING

KENNETH H. CONDIT, *Editor*

MCGRAW-HILL PUBLISHING CO., INC. JANUARY 1935 VOLUME SIX, NUMBER ONE

World Affairs

AS WE enter a new calendar year it is but natural to look about us to see how much the future will differ from the past, and how those differences will affect us. We product engineers are subject individually to these portending changes, of course, but professionally they are even more significant to us, because the things we design must meet the needs of a changing civilization.

Most of us are prone to believe that political events are of minor interest in our technical work, but the belief is a mistaken one. The present concentration of power in the Executive makes for drastic and rapid changes that will certainly modify the markets for our products. It is vital that we watch them closely.

For instance, the President's vigorous drive to get electric power costs down may well widen the market for domestic machinery, and it may also narrow the market for luxuries formerly bought by large holders of public utility securities if the immediate effect is to devalue these securities.

Likewise, the continued existence of a group, containing one-sixth of the population, which is out of work and is becoming accustomed to living off the other five-sixths will certainly lower the living standards of all. It is a tragic fact that the unemployed drift all too rapidly into the ranks of the unemployable.

Unemployment insurance, old age pensions, health insurance and like social reforms will inevitably change our living habits, and therefore the markets for engineered products. All of these reforms will be pressed in the new Congress. Payment of the soldiers' bonus and passage of a thirty-hour week law are both possible.

All these forces, and many others, act upon the markets for the things the product engineer designs. He must watch them constantly because he must make his plans far ahead if they are ever to get into production in time to be sold.

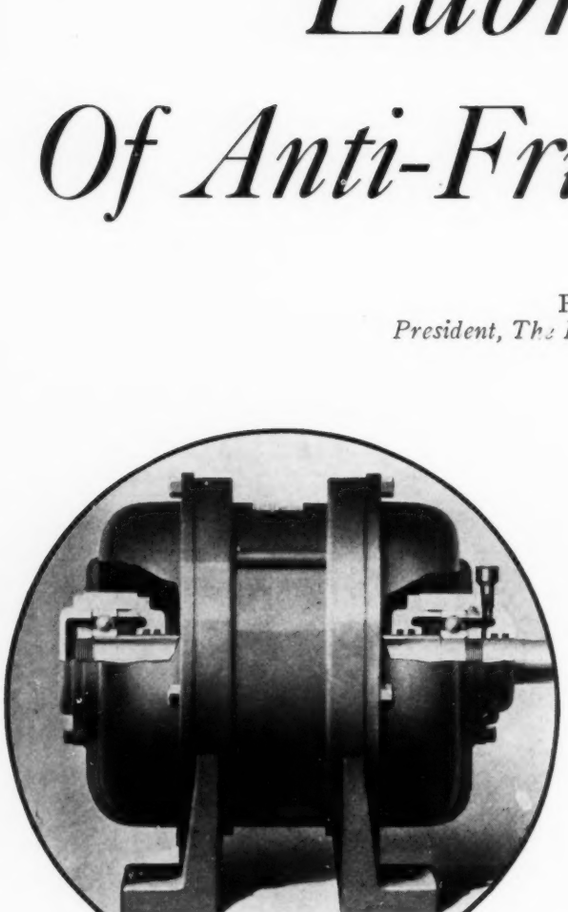
The country, fortunately, is busier than it was a year ago. More people have money, and they are spending more of it in the stores. Industry will almost certainly invest money in plant improvement this year. Our markets, therefore, will be larger even if they are harder to analyze, and the opportunity for product engineers is thus better than it has been for five years. Let us grasp it.

We Commence Volume VI

With this issue of *Product Engineering* we commence our sixth volume. The depression which has oppressed all of us for the past five years has made it a task to bring the paper through its teething stage, but its steady growth in acceptance places it in a favorable position to increase in usefulness to its readers as the country fights its way upward toward recovery. During the coming year the following augmented staff will edit it: Kenneth H. Condit, editor; George F. Nordenholt, managing editor; Robert W. Carson and Fred A. Firnhaber, assistant editors; Paul Wooton, Washington; Leonard Church, Cleveland; N. J. Van Ness.

Lubrication Of Anti-Friction Bearings

F. LEISTER
President, The Bearing Appliance Company



Motor windings will be damaged by oil; grease lubrication is preferred and much attention is given to sealing

Part I
Oil or Grease?

In this series of four articles the author, until recently chief engineer of the Fafnir Bearing Company, draws on his experiences of 15 years in the services of that company

OIL or grease? This question should be answered in the designer's mind during his preliminary design using anti-friction bearings—whether ball or roller type. True, many appliances present no problem in selecting the type of lubrication, but others require a study of all the factors to insure satisfactory bearing performance. Such study will often avoid costly changes and insure the purchaser and user the best lubricant for the given machine. But the decision to use oil or grease should not be based solely on the specific qualities of the lubricants but rather on what those qualities offer in combination with other conditions. Grease in a vacuum cleaner has the same advantages as grease in a 500 hp. motor, but other considerations may make it a poor choice for one or the other. The type of lubricant chosen should be that which will present the fewest disadvantages under the operating conditions.

No one knows the machine quite as well as its designer, and he is best fitted to determine its aim and the means for gaining that end. The service into which a machine is to go and the designer's provisions for meeting the particulars of that service, determine in some degree what shall be chosen as a lubricant for the anti-friction bearings. Conditions of temperature, humidity, air velocities, speed, accessibility, cleanliness, fire hazard and safety must be checked in connection with the selection of the form of lubricant. Utilization of a machine for service other than that for which it was designed originally, may suggest investigating service factors that are not present when the machine is used as originally intended.

With the service factors studied, the designer must then weigh the limitations of his design—first cost, available space, planes of shafts, and other peculiarities of the machine. Design factors may be fewer, but are just as important as service factors.

Operating Temperatures

While most apparatus operates at normal temperatures, which present no problem in lubrication, operating temperature is nevertheless an important service factor in the selection of lubricants for ball and roller bearings. For temperatures up to about 150 deg. F.

either grease or oil might be used. The oil may have a wide range of viscosity, or if grease is used, the kind of soap base or consistency may vary considerably.

On the other hand, extremely low or high temperatures may possibly demand oil because heat will dry grease and leave a hard, non-lubricating soap. Furthermore, high melting point greases of the soda base variety are soluble in water and therefore not suitable where moisture as well as heat is present. The insoluble greases, lime bases, have lower melting points and therefore are not usable for high temperature work. If, with high or low temperatures, other conditions demand that grease be used, then it is important to select a grease that will have a satisfactory consistency at the operating temperature—not the normal temperature.

While special greases have sometimes performed well at abnormal temperatures, oil is in general to be preferred since it has fewer disadvantages under such conditions. Nevertheless, its viscosity at the operating temperature is important and the oil selected should exhibit the least possible tendency to carbonize or vaporize. Viscosity decreases as temperature increases. Hence a light oil may become too thin at the operating temperature. At or near the safe maximum operating temperatures for anti-friction bearings, about 250 deg. F., the heavier mineral oils should be used if other conditions permit.

Oils, when refrigerated, become greases of vaseline-like consistency, and will not flow. Therefore, low cold-test oils are required for extremely low temperatures.

Moisture

When no moisture conditions need be considered the designer has wide latitude in choosing a lubricant. Greases of either lime or soda base will be satisfactory, although the latter may be preferable due to their greater stability. Oils of varying viscosities will also perform satisfactorily.

However, splashing water, or even extreme dampness, requires the finely finished surfaces of ball and roller bearings to be protected. Grease adhering to the bearing surfaces offers this protection during operation as well as through periods of idleness. But the lubricant must not be such that it will mix with water. This requirement excludes the soda base greases which are soluble. The lime base greases are therefore usually used for wet or moist conditions, and are superior to oil for this purpose.

A quick check for solubility is easily made with a small quantity of grease immersed in water in a small receptacle or in the palm of the hand. Insoluble grease will repel the water while a soluble grease will mix and emulsify, clouding the water.

Heavy oils will adhere to the polished surfaces of bearings during periods of inoperation. Mineral oils will not emulsify with water and are not affected by steam. Oils exhibiting any tendency to emulsify are unsuitable for ball or roller bearings. If because of other conditions oil must be used, the heavier the oil the better, as this will tend to offer the tenaciousness of grease. Light oils, unless demanded by some more important considerations, should not be used. For wet conditions, good lime-base greases are the best.

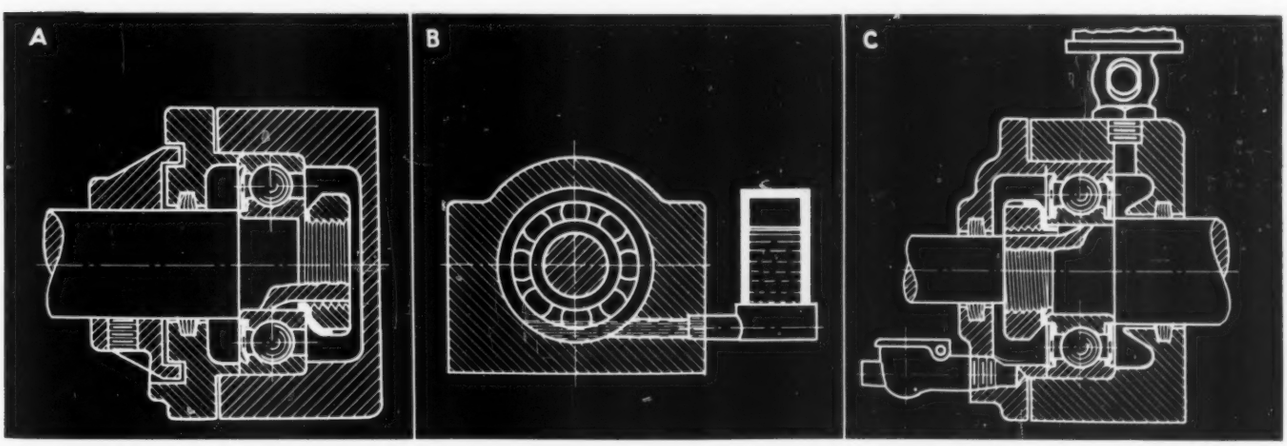
Windage

All rotating members cause air currents but ordinarily windage will not affect the selection of a lubricant. Normal windage, indeed, is commonly employed to circulate oil vapor through the bearings. Any air blast of unusual velocity, however, especially if coexistent with elevated temperature, might cause rapid vaporization of oil or the drying of grease. Grease is less adversely affected by air pressure and velocity and may be used efficiently when other factors do not demand the use of oil. Where oil is required, adequate seals, baffles, or in extreme cases, pressure equalizing devices, must be used.

Speed

Among the service factors, speed of shafts ranks close to temperature as an important factor in the selection of the lubricant. With the proper regard for other details, for low speeds the designer can use a great range

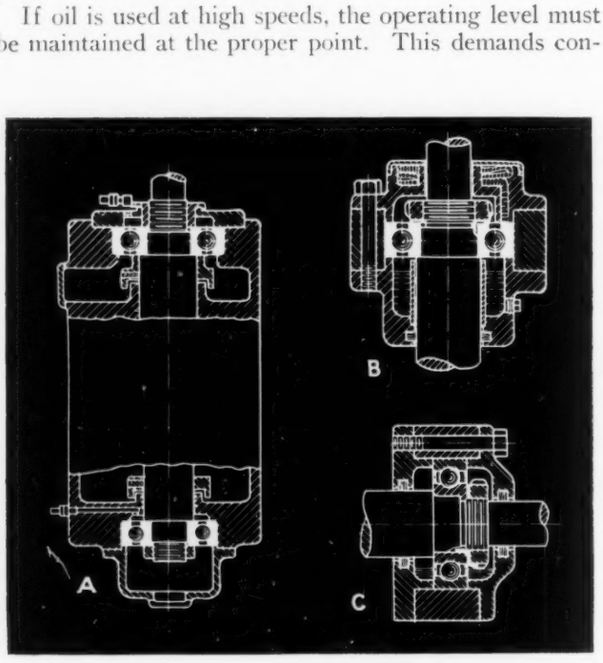
The finely finished bearing surfaces must be protected, as at ***A****, against the entrance of water. At* ***B*** *is shown one method of maintaining constant oil level, while at* ***C*** *is shown an oil-lubrication housing design for high speeds*



of oil viscosities or grease consistencies. Almost any system of oil supply or grease application will operate satisfactorily where speeds range only up to 500 r.p.m. Oil levels are not important and grease may be applied in a variety of ways and quantities.

On the other hand, high speeds present problems of lubrication. There are many important details. Viscosities, levels, consistencies, methods of feed or application, and venting of housings are only some of the matters to be considered. Contrary to the demands at low speeds, viscosities of oils and consistencies of grease are important. Oil level or the method of applying grease will also have its effects.

If oil is used at high speeds, the operating level must be maintained at the proper point. This demands con-



Grease lubrication on vertical shafts as at ***A*** *may require a bottom housing cover, or, as at* ***B*** *may involve careful provisions to make them oil tight. For oil-lubricated horizontal shafts as at* ***C****, the oil reservoir should be shallow*

trolling devices which may range from simple oil cups to elaborate circulating systems. Various types of oil-feeding and level-maintaining devices may have peculiar needs as to viscosity of the oil used. Heavy oils may not feed correctly through drip devices, as for example, through felt plugs such as used on some high-speed woodworking machinery. Light oil used in bath lubrication systems may be expended rapidly. Oil reservoirs should be large in area and shallow in depth, since the depth range is small in anti-friction bearings and the extra oil supply must be obtained by volume gained by area rather than depth. High oil levels will cause heating by churning. Housings may require venting to prevent expansion of air heated during operation which must escape from the confinement of the housing. Under some conditions it may be desirable to vent grease-lubricated housings, although this is rarely necessary.

Grease used at high speeds needs careful selection in order that its consistency be such that it will not definitely channel and yet be stiff enough to prevent heating from excessive churning. If the grease is too light it will be thrown off the fast-rotating parts, thus causing a deficiency. Because of these considerations the problem of lubricating high-speed bearings is two-fold: (1) Proper provisions must be made to apply and feed the lubricant in the desired amount; (2) The lubricants must be selected as regards oil viscosities or grease consistencies.

Importance of Continuity

In some applications more than the usual responsibility is placed on particular units. For example, a motor driving a machine tool is performing important duty, but a lubricant failure would shut down only the one machine. But a motor driving a train control system would, in the case of a lubricant failure, tie up a number of trains. Hence, the most efficient lubricant is demanded for vital apparatus. Extremes of temperature may be included in a duty study of apparatus operating out of doors. A small signal relay, for instance, may be in service in subtropical regions or it may be in far northern latitudes. One type of lubricant may be suitable for one location but not for another, and hence a wide field of application may require a study of all the probable fields of use. In low-torque apparatus, the selection of a lubricant is important in order that the one finally selected does not unduly affect the starting or operation of delicate apparatus.

With important units, grease has the drawback of being difficult to inspect as to quantity, either in the housings or while being added, unless covers are removed. Oil levels, on the other hand, can be observed continuously by means of the usual appliances. Oils, where used for high-load apparatus, may be of the heavy varieties and assist in reducing shock loads on retainers and other bearing parts. Some duty requires bearings to be provided with heavy piloted retainers, and these require oil lubrication. Also, it may be desirable to lubricate angular-contact bearings with oil, since these types, because of their construction, are inherently liable to generate more heat.

In extremely vital apparatus, oil circulating systems may be required. Continuity of use in important apparatus, it is seen, may demand the most efficient lubricant—oil—sometimes attended with adequate level apparatus or circulating devices. And users of important apparatus should not change from one type of lubricant to another without consulting the manufacturer of the equipment.

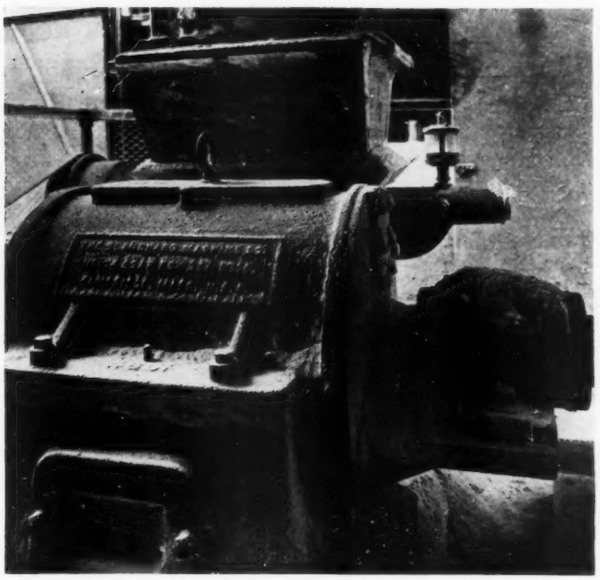
Accessibility

Whether or not bearings are accessible has, furthermore, a relation to the selection of grease or oil in anti-friction work. Oil is a mobile lubricant, and will flow to the point required, while grease must be forced mechanically to the point of need. Remote oil cups are likely to be overlooked, and therefore wherever possible inaccessible bearings should have large oil reservoirs. Small quantities of oil may be consumed rapidly and small quantities of grease may dry or harden. Both oil and grease are easy to apply, but oil may have to be renewed oftener than grease. Some ball bearing

applications have run a number of years without adding to the original grease.

The number of bearings to be lubricated is another factor, as the number of oil cups needing frequent refilling may constitute a maintenance item of some size. Where there are many points to be lubricated and where there is a variety of conditions—heat, moisture, etc.—it may be well to consider oil, since a greater variety of conditions can be met with fewer oils than can be met with greases. For example, if a high melting point grease is needed at some point on the machine and a non-soluble grease is needed at another, at least two greases will be required. But, if the designer arranges for it at the outset possibly one oil will lubricate all bearings.

Here may be mentioned the possibility of the same machine being used in widely different points, such as both tropical and sub-arctic regions. Oil is easy to get almost anywhere, while particular, specific greases may not be available in remote regions. The time between



Machines that must operate under conditions such as this must have the bearings well protected against dust or dirt

lubricant renewals, accessibility of bearings, size of reservoirs, safety of operators, number of bearings to be maintained, remoteness of apparatus, and availability of lubricants have to be considered in choosing between oil and grease.

Cleanliness

In food, textile, paper and other industries no oil drip or excess grease can be tolerated. Further, even in industries not so rigid in this regard, oil on belts, clutch facings and brakes must be guarded against. Electrical windings must be protected from oil for two reasons: first, windings will deteriorate badly if oil soaked, and second, oil on the windings will permit dust to accumulate until important air passages become restricted. Either damaged insulation or impaired ventilation may lead to burn-outs. Grease is considered the cleaner, and with modern equipment can be kept away from products and machine elements needing protection. Where the danger is extreme, even grease-lubricated housings may be sealed with gaskets between the main housing and cover, especially if any separation of the grease is likely, or if elevated temperatures are to be experienced. Oil is less clean to handle, as it drips and spatters and may ruin the product of the machine if it comes in contact with it. Mineral oils, especially if sunlight is present, will permanently stain fabrics.

Protection from Dirt

Not only must the product of the machine be protected, as in the instances mentioned above, but in some other applications the bearings themselves must be protected against dirt, grit, acid, or similar conditions. Dirt, mixed with grease, will rapidly wear out ball and roller bearings. A mixture of grease and dirt is almost impossible to remove from a bearing housing without completely disassembling it. If it is expected that frequent cleaning will be necessary, cross feeds and drains —that is, inlets at the top of the bearing housing and outlets at the bottom—may be desirable. Grease assists in sealing out dirt by packing around shaft openings. While oil does not offer this advantage, dirt will settle out of it into well-designed sumps, or in circulating systems, and can be filtered out. Oil will wash dirt off bearing surfaces, but will run off the bearing parts and seals during idleness. In extreme cases of dirty atmospheres, stuffing boxes may be required to exclude damaging dust, liquids or grit particles.

Fire Hazard

In apparatus such as ovens and furnaces, care must be taken to eliminate fire hazard from the fumes of lubricating oil. When grease cannot be used, oils with high flash and fire points should be specified and provisions made so that the machine can be oiled easily without excess oil being spilled over floors, platforms or machines, and thereby avoid serious consequences if an open flame is brought near the oil-covered surfaces. While anti-friction bearings have been instrumental in reducing fire hazards around machinery, oils should be handled carefully around any apparatus involving high temperatures or open flames.

Design Factors

All the foregoing service factors are worth studying in some designs, and some are worth thought in all designs. Peculiar service conditions may alter a selection by virtue of one or two exceptionally important features in a specific case. Design factors, while not so numerous, are nevertheless just as important, and require thought on the part of the machine designer.

In the next article, which will appear in ***February****, the author will discuss the peculiarities of* ***oil and grease*** *as lubricants. Results of tests and other* ***investigations*** *are included.*

PRODUCT DEVELOPMENTS

Inertia Forces in a Bread Slicer

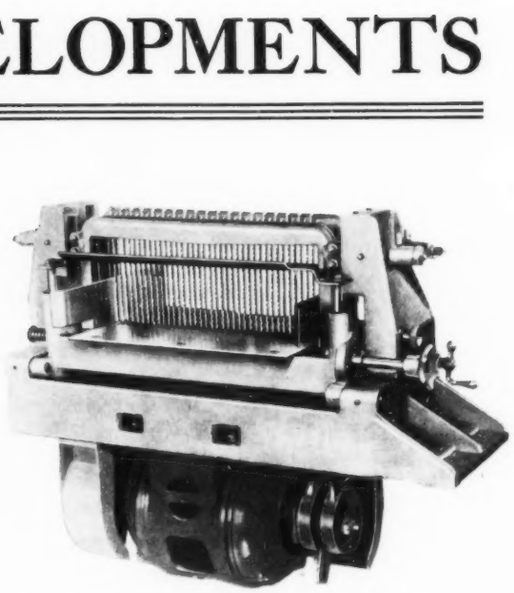


Fig. 2 — Light-weight connecting rods of high strength alloy steel reduce inertia forces in the oscillating knife mechanism

IN THE development of a machine built by the American Machine & Foundry Company, for slicing and wrapping loaves of bread, high-speed operation involved several problems in inertia forces. In this machine, shown in Fig. 1, loaves are fed into the lower level, and after passing through the slicer are lifted to the upper level, where the wrapper is applied. When unsliced loaves are to be wrapped, the slicing unit is withdrawn from the machine.

To permit this unit construction, the slicer has its own drive motor with control interlocked with the wrapper mechanism. For high-speed slicing, two knife frames are oscillated 1,800 times per minute by counterbalanced cranks mounted on each end of the motor shaft, as shown in Fig. 2. The high inertia forces involved presented several problems in arranging for a compact, self-contained slicing unit.

Aluminum alloy was used for the knife frames, but the low stiffness of aluminum made it necessary to supply individual tension springs for each blade. With 60-lb. tension on each blade, the total load on each of the two frames is nearly a ton, requiring a carefully designed frame section to keep working stresses within safe limits.

For rods connecting the frames to the motor, aluminum alloy was desirable to reduce inertia forces, but ball bearing outer races pressed into aluminum rods tended to work loose, the bearing then rotating within the housing. Heat-treated, chrome - molybdenum steel rods were therefore used and were found to be both stiff and light in weight, holding the bearings tightly in position.

Although a standard 1½-hp. motor supplied enough power to drive the knife frames, the motor shaft and bearings would not carry the high pulsating stresses set up by the inertia forces. By substituting a chrome-nickel steel shaft of the same diameter, carried on roller bearings, it was possible to use a standard motor frame.

Inertia loads were also a factor in the design of the lifter table which transfers loaves from the

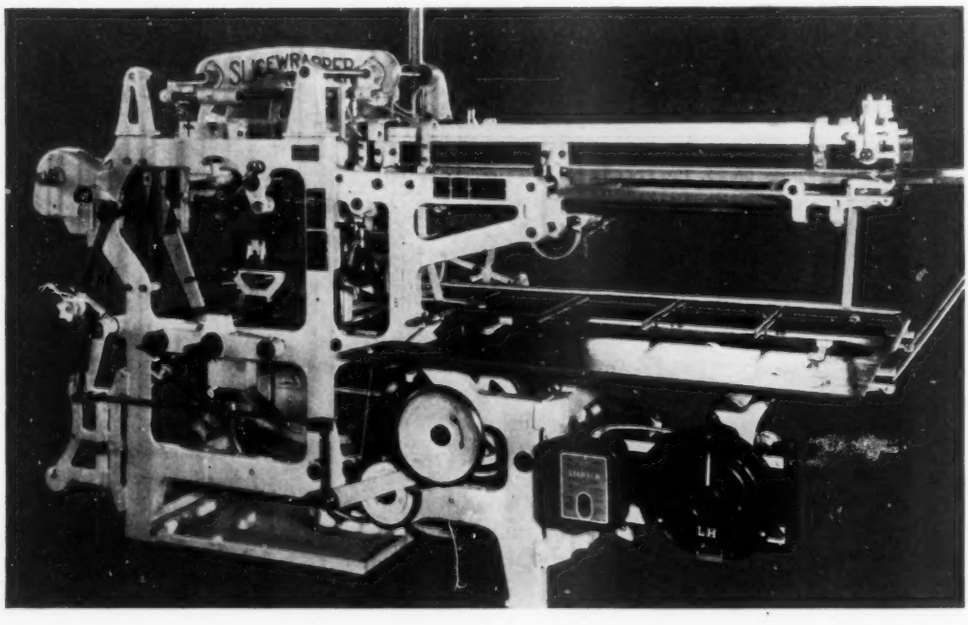


Fig. 1 — Almost a catalog of modern materials and parts, the Slicewrapper incorporates: built-in illumination; magnesium alloy and aluminum alloy parts for less inertia; alloy steels for highly stressed parts; ball, roller, needle, sleeve and oilless bearings as required; spot heaters, built-in wiring, variable speed drive, safety controls

slicing to the wrapping level, the lifter table traveling 12 in. in less than ½ sec. To reduce bearing loads and power requirements, this table casting was made of magnesium, resulting in a table much stiffer than if made of sheet metal of equal weight. Another reciprocating mechanism, a pusher that discharges the wrapped loaves, takes its power from an oscillating chain drive which has less inertia and therefore requires less power than a linkage mechanism. Since the pusher plate passes over the loaves, oilless bearings are used to avoid possible damage from dripping lubricant.

Quiet operation is secured by Micarta pinions meshing with helical gears in the wrapping machine drive, while rubber is used extensively for bumpers and stops. A long steel bar that gave an objectionable ring was silenced by wrapping around it a helical spring.

Careful attention to electrical wiring resulted in inclosing nearly all the wires in cored channels in the frame or in built-in pipe conduits. Connections for spot heaters and the slicer motor are readily disconnected without disturbing the wiring, and the built-in light serves as a pilot light in addition to illuminating the lower table. Although an across-the-line switch would be suitable for the wrapper drive, a magnetic starter allows the use of safety switches to prevent the motor from starting when the slicer is not in its proper position.

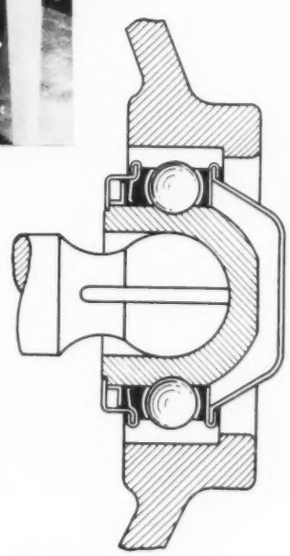
Forming the central feature of the dining alcove, this table is made entirely of glass disks and cylinders. Plywood walls have a baseboard strip of Micarta

• *Sealed Bearings for Conveyor Rolls*

Water, sand, cement dust or other injurious substances must be excluded from ball bearings used for conveyor rolls, and because of the large number of bearings in the usual installation, lubrication must require a minimum amount of attention. Also, to confine rotation to the bearing itself, the design of the mounting must prevent relative movement between the bearing races and the roll or frame. Easy assembly and removal of the bearings is also desirable when rolls are subject to considerable wear and replacement is a factor.

To meet these several requirements, a new bearing shown in the accompanying diagram was developed by the New Departure Manufacturing Company. The bearing is designed around a special inner race which has a spherical pocket taking a slotted stub shaft which is locked into the frame. The outer bearing race is a press fit in the conveyor roll, and can be removed readily from worn rolls.

With the bearing permanently sealed and lubricated for life, foreign matter is excluded, and pressure gun fittings or oil lubricators are eliminated. The design also makes adjustments unnecessary, while the slotted end on the stub shaft fitting tightly into the inner race prevents rotation and consequent wear between the inner race and the shaft.



Designed especially for conveyor rolls, this New Departure bearing is permanently sealed and lubricated

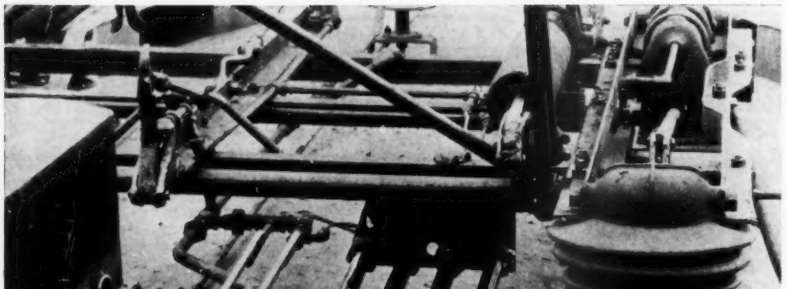
• *Furniture of Glass*

Unusual design effects are obtained by extensive use of glass in a dining alcove shown in the illustration to the left. The table designed by Walter Dorwin Teague is entirely of plate glass. The table top is a heavy disk supported on four cylinders, these in turn resting on a double base of mirrored plate glass and black glass.

This table is the central piece in a circular alcove in which the walls are made of plywood with a Micarta baseboard. The narrow window, at eye level height to persons seated at the table, is screened with vertical glass rods.

• *Modernizing Elevator Controls*

Hazards present in the operation of the shipper rope in cable-controlled elevators, particularly when calling the elevator from a landing floor, are reduced by the application



Air lines operating pantographs on Westinghouse electric locomotives are insulated by porcelain tubes sealed to aluminum connections with rubber bushings. The seal is designed so that air line pressure increases the effectiveness of the seal

of a torque motor for shifting the elevator control cable.

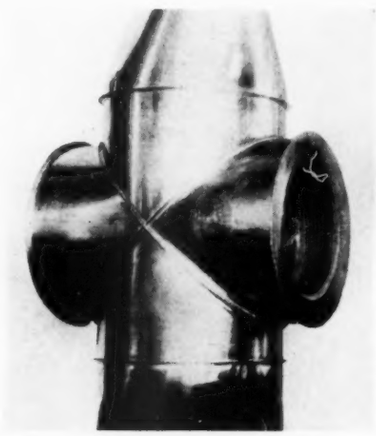
In the control system worked out by the Quincy Elevator Gate Company, a reversible torque motor moves the shipper rope through a reduction-gear drive, operation of the motor being controlled by "up" and "down" electrical contacts at each floor and in the car.

Special windings for the motor allow it to stall on the line without injury, and a built-in control device gives slow speed at each travel limit, which made it unnecessary to provide limit switches.

• *Welding Tantalum*

Tantalum sheets possess an unusual combination of ductility and resistance to wet corrosion, characteristics leading to extensive use of this metal in chemical equipment and laboratory accessories. But the welding of tantalum presents several difficulties. Although the melting point is 5,100 deg. F., tantalum sheets lose ductility when heated above 650 deg. F., so that welds made by usual methods are brittle.

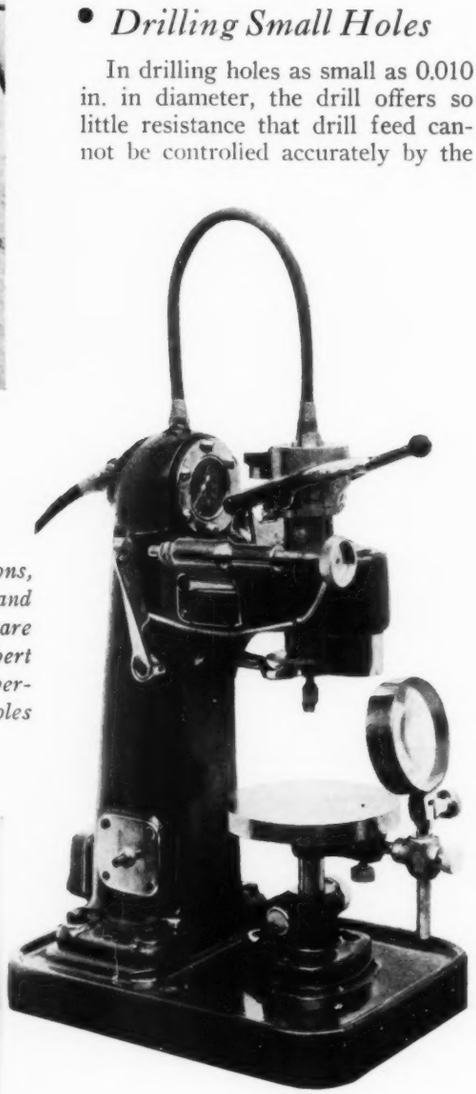
Using a high-voltage, low-current welder made by the Lincoln Electric Company, sheet tantalum is now being welded successfully by the Fansteel Products Company by immersing the parts in carbon tetrachloride. Welding is accomplished by melting the edges of flanges formed at the joints, striking the arc but once at each place. Sheets less than 0.02 in. thick are being welded by this process, greatly increasing the field of application for this metal.



Chemical equipment made of sheet tantalum is arc welded while immersed in carbon tetrachloride to prevent overheating

• *Drilling Small Holes*

In drilling holes as small as 0.010 in. in diameter, the drill offers so little resistance that drill feed cannot be controlled accurately by the feel of the feed lever or by the operator's sense of touch, also, exceptionally good vision is needed in centering such small drills over the work. The small drill diameter also requires extremely high rotational speeds to obtain proper cutting speeds. And, with such small drills, vibration must be eliminated as far as possible.



Because of human limitations, a built-in magnifying lens and an unusual feed mechanism are incorporated in the Herbert sensitive drill to assist operators in drilling small holes

In the Herbert sensitive drilling machine shown here, the spindle runs at speeds of 60,000 to 80,000 r.p.m. Drive is by an air motor operating under a pressure of 80 lb. per sq. in. The speed, controlled

by a throttle valve, is indicated on a dial at the top of the column.

For controlling feed, pressure on the hand lever moves a second lever at the top of the machine into contact with a stop, bringing the drill point down to the work. While pressure is still maintained on the hand lever, the stop is slowly withdrawn by screwing out the knurled hand wheel. From the sound of the drill as it penetrates the work, the operator can determine the correct feed, too fast a feed causing a fall in pitch.

After the hole is drilled, the stop is screwed in to lift the drill clear of the work, and the drill spindle is returned to normal position by releasing the control lever. A spring-loaded fork bears against a flange on the drill spindle, raising the spindle when the control lever is released by the operator.

Because of the limitations of human vision, a magnifying glass is designed into the machine to make certain that every operator can see what he is doing with even the smallest drill. By selecting a large diameter glass, work can be viewed with both eyes, thus reducing eye fatigue.

• *Over-Running Clutch Refrigerator Drive*

In the new 1935 model General Electric refrigerated truck body a specially designed power take-off has an over-running clutch disconnecting the generator armature when the gears are being shifted. By disconnecting the high-inertia armature, the gearing slows down in a normal manner when shifting.

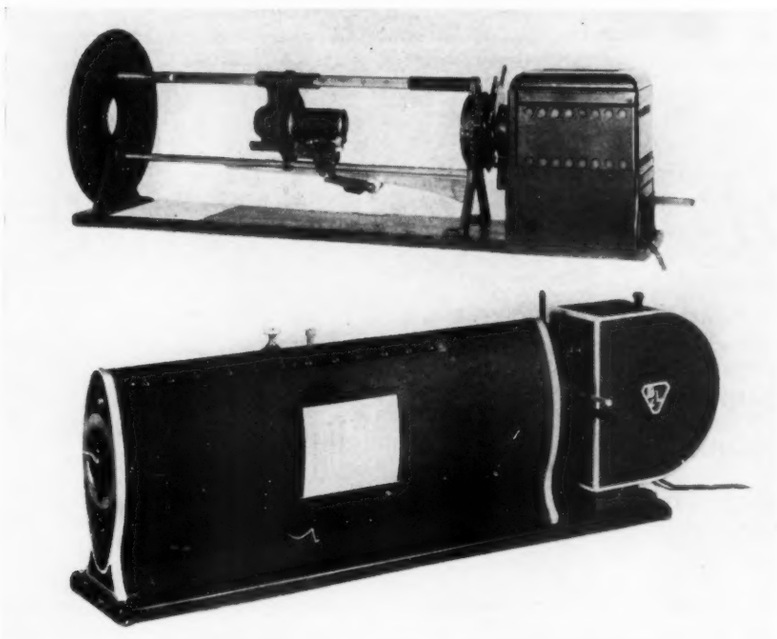
The over-running clutch is built into a driving pulley connected to the generator by a multiple V-belt. The generator is in the machinery compartment at the front of the truck body, the drive being protected from mud and dust by a metal housing.

Double ball bearing journal box and two needle bearing universal joints carry the drive shaft which transfers power from a take-off on the transmission to the drive pulley. A standard S.A.E. opening in the transmission case accommodates the power take-off.

• *Good Appearance for First Impressions*

Instead of the familiar illuminated charts used in testing vision, the Clason Acuity Meter built by the Bausch & Lomb Optical Company projects sharp images on a screen, the image size being controlled by moving the optical slide on the projector. As first designed, this instrument was entirely satisfactory in performance, measuring keenness of visual perception with greater accuracy than obtained with the usual card method. But, as shown in the upper view no attempt was made to "dress up" the instrument.

As ordinarily used, the instrument occupies a prominent position among the optical apparatus used in eye-testing, and for this reason, the appearance of the meter is a factor in establishnig the patient's mental appraisal of the practitioner's skill. A redesigned instrument, shown in the lower illustration, using the same elements and the same physical dimensions, made possible a more impressive piece of equipment.



Although embodying the same essential elements as the first model (above), the redesigned eye-examining instrument (below) creates a more favorable impression and has less tendency to distract attention

Designed by the West Bend Aluminum Company, this cocktail shaker consists of a drawn copper shell, chromium plated, with Bakelite top and bottom

Design for Better Seeing

Built-in lighting equipment giving scientifically designed illumination reduces human errors and increases production. Leaders are offering such lighting — others must follow

ATTENTION now being given to general factory lighting is bringing greater recognition of the advantages obtained from adequate illumination. Production managers and factory executives are becoming "foot-candle" conscious because they are finding good lighting profitable. But it appears that most designers of machinery still leave it to the operator of the machine to "rig up" his own lighting equipment, or compel the shop maintenance department to provide a makeshift arrangement. In either case, usually no attention is given to the illumination requirements, to intensity of illumination, type of reflector, direction of the light beam, and the other factors that enter into a lighting problem.

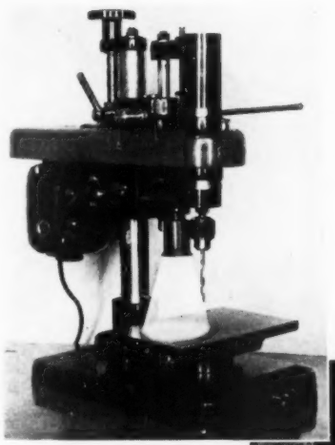
In some shops where the value of adequate local light is appreciated, an individual lamp and reflector is attached to the machine or swung from a convenient support. But, more often, the light is supplied by an ordinary open lamp on a drop cord, the lamp usually being tied to the most convenient place that the operator can find, with no regard to illumination efficiency.

In improvised illumination of the type shown above at the left, glare places a severe strain on the operator's eyes, while excessive contrast in light reflected from the work actually decreases vision. Light from the lamp is scattered instead of directed where it is needed. But good illumination, as shown at the right, shields the eyes from the high intensity light from the lamp. The reflector also gives more uniform distribution of light, eliminating shadows, and, in addition, the bulb is protected against breakage. The lamp cord being inclosed in a flexible metal tube protects the cable from wear, eliminating the danger of short circuits. With the flexible support, the lamp can be swung out of the way if necessary, yet it can be placed so as to throw the most light on the work.

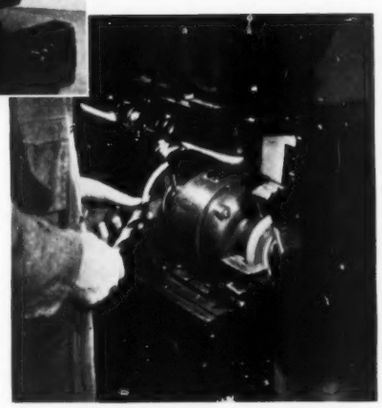
A few scattered designers have "cashed in" on the increased machine performance and added sales value contributed by well-designed built-in lighting. These designs, such as the sensitive drilling machine and the tool grinder shown on the next page, insure good lighting of the work with minimum eye fatigue for the

operator. In addition to the few American machine tool builders, several European machine tool manufacturers offer complete lines of machines with built-in lighting specially designed for the job. These designers know that the performance of their machines will not be handicapped by "poor seeing."

Designers in other fields seem to be more aware of the advantage of "good seeing," as shown by the constantly increasing use of built-in lighting in business machines, household appliances, wrapping and packaging machines and optical equipment. The group shown at the bottom of the page shows several applications where light, properly directed and diffused, reduces human errors, speeds production and adds convenience.



Poor lighting will never penalize the performance of the Muehlmatt sensitive drilling machine



A tool grinder needs plenty of light. This designer took no chances with improvised lighting

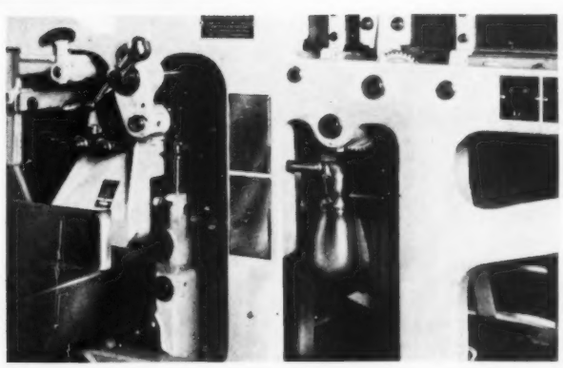
When the new Hoover vacuum cleaner first appeared, many looked upon the "headlight" as a novelty. But it is significant that women now demand this added convenience, that the built-in light is not merely a selling feature, but an important factor in increasing the utility of the cleaner. The obvious advantage of having light always directed in the path of the cleaner has thus been an important factor in its sales success.

Greater speed and fewer errors have resulted from built-in lighting on business machines. Adequate illumination properly directed not only relieves eyestrain but also gives more accurate vision, making possible greater performance with the machine. Sewing machines for many years have been sold with built-in lighting as standard equipment.

In optical equipment such as microscopes well directed light of the proper intensity is assured by the use of scientifically designed integral lighting units. In the bread slicing and wrapping machine designed by the American Machine & Foundry Company an inexpensive fixture, shown below, throws a flood of light over the loaves of bread as they pass through the machine. Better illumination results directly in fewer spoiled loaves, and makes possible a higher operating speed. Cash registers have long used built-in lighting for the obvious advantage of avoiding mistakes, and electric refrigerators now have lights switched on when the door opens.

In each of these typical examples, needed light is supplied by built-in equipment designed to give the best illumination. In each instance the character of the illumination furnished is based on a study of the lighting requirements and the desired lighting is achieved by correct mechanical design with reference to the lighting equipment used and its mounting or attachment.

Illumination obtained from properly designed integral lighting equipment supplies better seeing than the machine user can obtain from attachments placed on the machine after installation. For only when good lighting is considered in the design can provisions be made to incorporate the necessary switches, electrical connections, reflectors, supports, etc., in the most economical



Many designers in other fields are already "cashing in" on the sales advantages of integral lighting, giving the user a quality of illumination unobtainable by makeshift attachments

and convenient manner. In many instances, proper illumination is not possible if it is not built-in. Thus, only the designer of the machine can build in a quality of illumination that the machine user cannot obtain otherwise except possibly at considerable expense. To the many machine buyers who appreciate the advantages of better seeing, the elimination of extension cords, swinging lights, unprotected lamps, short-circuit hazards and expensive installation costs are real sales advantages.

Many machine users, however, do not go to the expense of supplying proper local lighting when the designer fails to provide adequate illumination. Machine buyers or operators cannot be expected to make a study of the lighting requirements for each of their machines, selecting the proper lamp and reflector for the job, and finding a place on the machine or floor or wall for the light support. As a consequence, the machines ordinarily are used with whatever room light is available plus that obtained from poorly improvised extension cords.

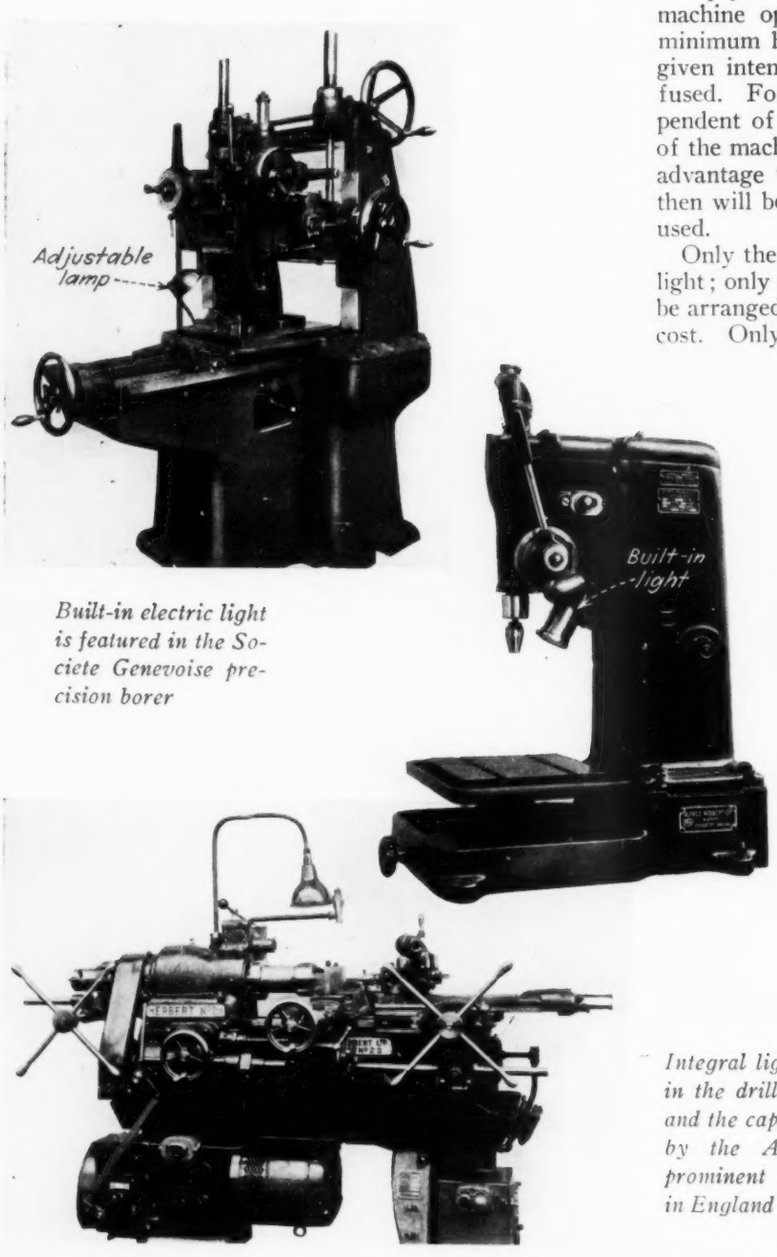
Lamps and reflectors used for general room lighting cannot supply the illumination needed for best vision in the working area of the machine. The machine operator may make the best of a poor situation by tying up a drop light to throw more light on the work. The operator may attempt to reduce glare from an unprotected light by tying a piece of paper around the lamp. Such methods never give adequate light, are inherently wasteful, and many times are actually harmful.

Only when designed as an integral part of the machine can local lighting be obtained with minimum current, lowest equipment cost, least wiring expense and of proper characteristics for maximum efficiency.

Those who have made a study of the physiological and psychological effects of lighting have shown that a machine operator will deliver maximum output with minimum human error only when illumination is of a given intensity, properly directed, distributed and diffused. For any given machine, these factors are independent of general room lighting, so that the designer of the machine can provide the best illumination to best advantage when designing the machine. The lighting then will be correct regardless of where the machine is used.

Only the designer can select the best position for the light; only in the design layout can the electrical wiring be arranged as an integral part of the machine at lowest cost. Only by providing the correct amount of light, directed and diffused as needed, can the designer insure that the operator of each of his machines will always have the best light for accurate and rapid seeing, accurate and rapid production.

It is not a difficult problem to build-in efficient illumination. Research in the general lighting field has already established requirements for maximum speed and accuracy in seeing, and equipment developed for other lighting problems can be used for machine illumination. These requirements for built-in illumination involve such factors as reflectivity of painted surfaces, color and intensity of illumination for maximum vision, glare, intensity contrast, and diffusion. These factors, and the illumination characteristics of various lamps and reflectors will be given in the February number.



Built-in electric light is featured in the Societe Genevoise precision borer

Integral lighting is incorporated in the drilling machine (above) and the capstan lathe (left) made by the Alfred Herbert Ltd., prominent machine tool makers in England

Stampings–II
Design of Drawn Parts

J. K. OLSEN
Formerly Chief Draftsman, Stewart Warner Company

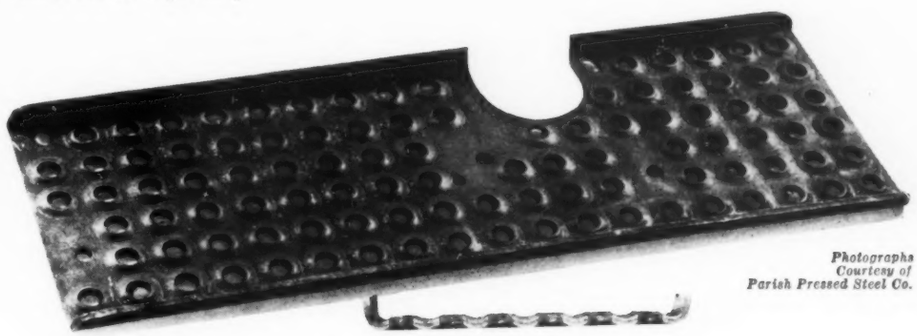
THE designing of drawn metal stampings such as shells and cases is interesting and has many possibilities but it also presents numerous problems. Not only must the designer know the basic rules for shaping strips and sheets into the forms required, but he must also know the limits for a practical design. There is a wide field for variations of design and a designer should try to avoid unnecessarily deep draws. Parts that are drawn extremely deep involve complicated operations, thereby increasing the cost of production.

In the production of drawn parts the method used depends on the depth of the draw and the kind of material. All parts that are to be drawn should be made of sheets sufficiently thick so that the body of the part will have the required thickness. Thin sheets may have a tendency to crack when given a deep draw.

Another factor that must be considered, especially when the part is to be held to close limits, is the commercial variation of stock thickness. Usually if the part is made from stock of maximum thickness it will be slightly longer than if made from stock of the minimum thickness, using the same size blank in both instances. Therefore, if accuracy is required it often becomes necessary to use material with small tolerances.

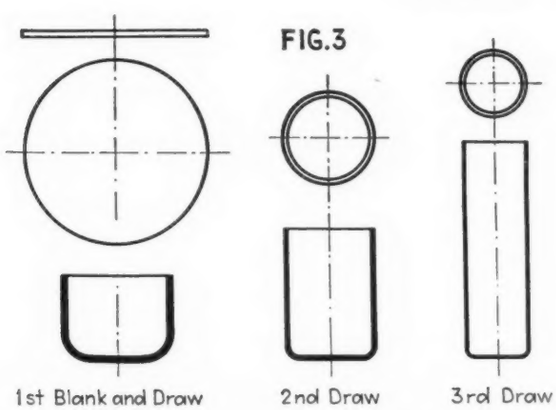
Fig. 1—Cup for 1 in. O.D. needle bearing race, the most common type of drawn shape

Fig. 2—Brake step for steel hopper cars, made of ¼-in. plate. Although the metal around the holes is punched up, the action is swaging rather than drawing



Photographs Courtesy of Parish Pressed Steel Co.

FIG.3



Probably the most common drawn shape is the cylindrical shell. Fig. 3 shows the various draws for such a shell, the approximate proportions of each draw, and the sequence of shapes and operations.

As can be seen from the illustration, when drawn parts such as shells and cases are made it is generally good practice to make the first drawing operation with corners rounded to a radius larger than the desired final radius. When designing parts with sharp corners, the material is liable to crack in the corners, thereby increasing cost because of the greater amount

of scrapped parts. Cracking of the material can often be avoided by annealing, but this also adds to the cost.

To obtain best results the kind of material should be considered and the designer should indicate well-rounded corners wherever permissible so as to give as much leeway as possible in forming and drawing, avoid annealing, and so as not to require material that must meet exacting physical requirements.

Often when designing drawn parts it is necessary to figure the weight of the part and have a definite thickness of the wall after the drawing operation. In most cases the stretch factor can be ignored because the stretch varies and generally is insignificant. As shown in Fig. 5, usually the wall thickness can be assumed to be the same as the gauge of stock used.

In Fig. 6 are shown parts that have been "ironed down" or upset intentionally, thereby decreasing or increasing thickness respectively. In parts of this type the difference in thickness due to additional operations must be considered. However, parts in this group are very few.

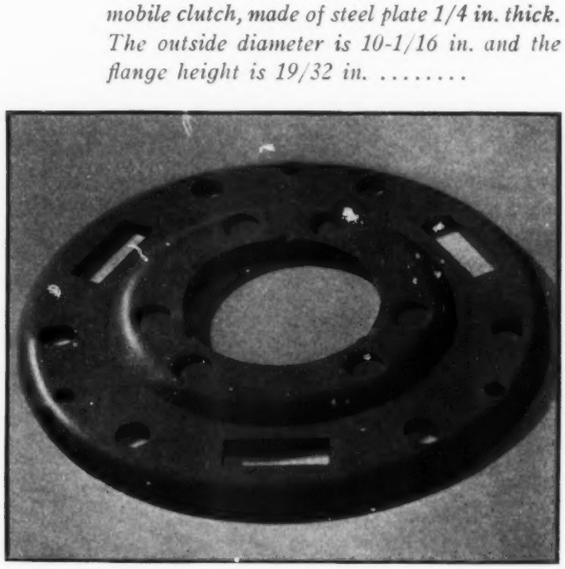
There are unlimited opportunities for obtaining best results in the forming of parts, but the engineer should always consider his design from the production angle. It is easy to incorporate various ideas that would cause excessive hardships both in the making of the dies and the manufacture of the part, thereby resulting in excessively high costs.

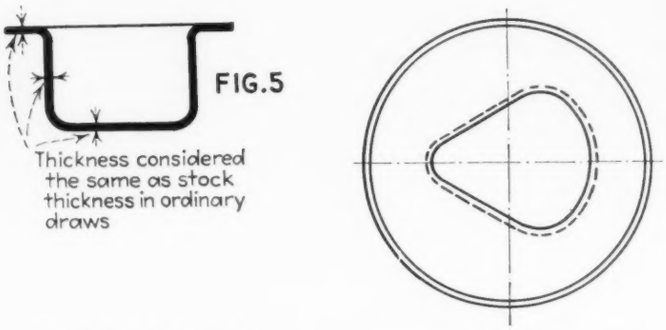
Besides sharp corners, there are many shapes that make it difficult for the metal to flow in the drawing operation. Fig. 7 illustrates a typical example of a part wherein a large radius should be allowed. A good stamping designer should be able to visualize the characteristics of drawing operations of this kind.

Many times the finished part does not indicate the method of manufacture that is to be followed to procure the final product. A typical example of this is the drawn shell shown in Fig. 8. Often the part is first made as a drawn cup, unless the part is made out of plain cylindrical tubing or tubing with a reduced portion. For example, the part shown in Fig. 8 would undoubtedly be made as indicated in Fig. 9. That is, the small diameter end of this part would at first be the bottom of a drawn cup and a hole punched in the bottom. The diameter of the hole would be of such size that when the final operation is performed the result would be a straight cylindrical section of the desired length and diameter.

Small drawn parts with a number of operations,

Fig. 4—Reaction plate for automatic automobile clutch, made of steel plate 1/4 in. thick. The outside diameter is 10-1/16 in. and the flange height is 19/32 in. .....





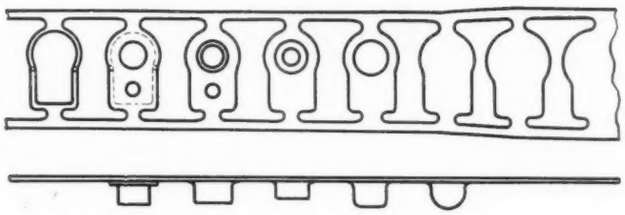


FIG.12

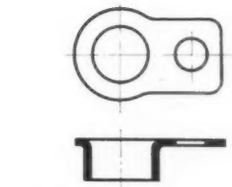


FIG.11

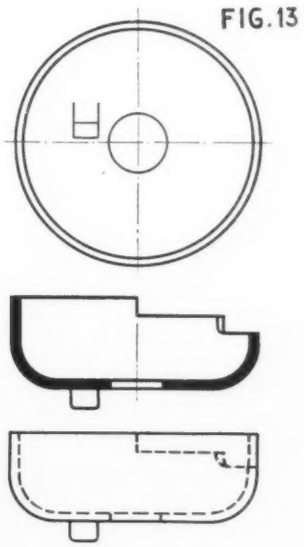
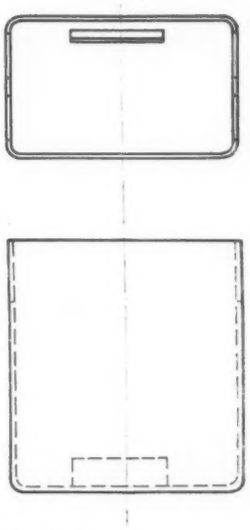


FIG.13



End View

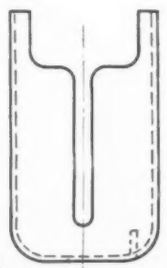
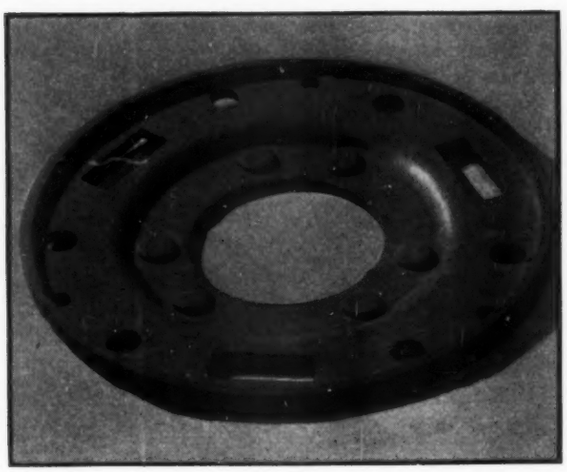


FIG.14

.....(Below) Reverse side of the reaction plate. This illustrates some of the possibilities in the design of stamped parts. The large central boss is a shallow draw



Courtesy Parish Pressed Steel Company

such as hubs, which can not be made in combination dies or single draws are often made in progressive dies, providing the quantity to be produced warrants it. This method varies from the ordinary method of handling drawn parts and has the advantage of producing a part that is complete when it comes off the die. It is an automatic step by step forming of metal into the required shape in a single die, regardless of the number of operations. To make such a die to produce the part in the least number of operations takes keen judgment and a great deal of practical experience.

Progressive dies usually have the disadvantage of requiring an increased amount of raw material. This is because of the practice of relieving the material between operations by removing a portion of it to allow the material to draw freely without the usual restriction of a solid strip.

Stamping the relief is generally the first operation. By adding this relief between blanks, as shown in Fig. 10, it can be seen that the multiple of the raw material strip increases correspondingly, but the convenience and speed of production that is gained will more than offset this increased cost of material.

To illustrate the method of making drawn parts with a progressive die, Fig. 11 shows a terminal with a drawn hub for spinning over in the assembly, as in assembling eyelets. It is interesting to note the changes through the various operations until the final one when the part is completed. As can be seen from Fig. 12, the method of drawing the embossing by gradually decreasing the diameter and increasing the length at the same time, is similar to the ordinary drawing of hubs.

The part shown in Fig. 13, or any part similar to it, probably would require operations as follows: (1) Blank and draw; (2) Clean; (3) Shear side; (4) Shear and form lug; (5) Perforate; (6) Clean.

A part similar to that shown in Fig. 14 probably would require operations as follows: (1) Blank and draw; (2) Clean; (3) Redraw; (4) Redraw; (5) Clean.

In the design of drawn parts it is necessary to calculate the size of blank required. There are several methods for doing this but because of the many variable factors involved, in many instances the only reliable way to determine the size of the blank accurately is the "cut-and-try" method. But often it is necessary to determine the size of the blank from the design of the part. Methods for doing this will be explained in the next article (*February P.E.*) of this series.

▼ ▼ ▼

Information to Give When O

ROBERT G. N. EVANS
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BEARING manufacturers are constantly in receipt of inquiries requesting them to recommend a bearing metal for a given size bearing and shaft, a given r.p.m. and a given load. Sometimes the information is more detailed but as a rule insufficient data are given to enable the manufacturer to calculate only the bearing dimensions, let alone to recommend a bronze composition that will give the best results. Many times the composition that has been used is stated and the bearing manufacturer is asked to duplicate it. This is of course all right in the event the bearing in use has been giving satisfactory service. But for the most part much data and information that would be of mutual benefit are entirely lacking and it is the purpose of this article to outline the many details that must be known in order that the correct bearing calculations and recommendations can be made and satisfactory performance assured.

In the accompanying Inquiry Form, page 21, all the factors pertinent to a bearing problem are listed and broken into several divisions to clarify the terms used and to indicate the necessity for complete information concerning a problem. The following explanations clarify the items in the inquiry form:

1—Load

The maximum load that is applied to the bearing is one of the factors that must be known before the bearing material, wall thickness and projected area can be determined. Loads are of three kinds or types:

(A) CONSTANT—By constant load it is meant that the

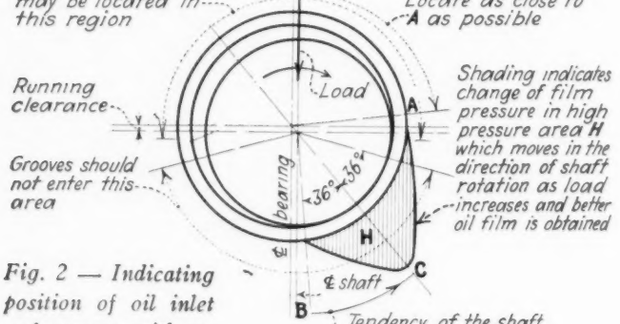

Longitudinal oil grooves may be located in this region
Area of oil inlet Locate as close to A as possible
Running clearance
Load
Shading indicates change of film pressure in high pressure area H which moves in the direction of shaft rotation as load increases and better oil film is obtained
Grooves should not enter this area
bearing
36°
36°
A
H
C
B
℄ shaft
Tendency of the shaft to climb causes the high pressure area H to move to the right from B to C

Fig. 2 — Indicating position of oil inlet and grooves with respect to the load

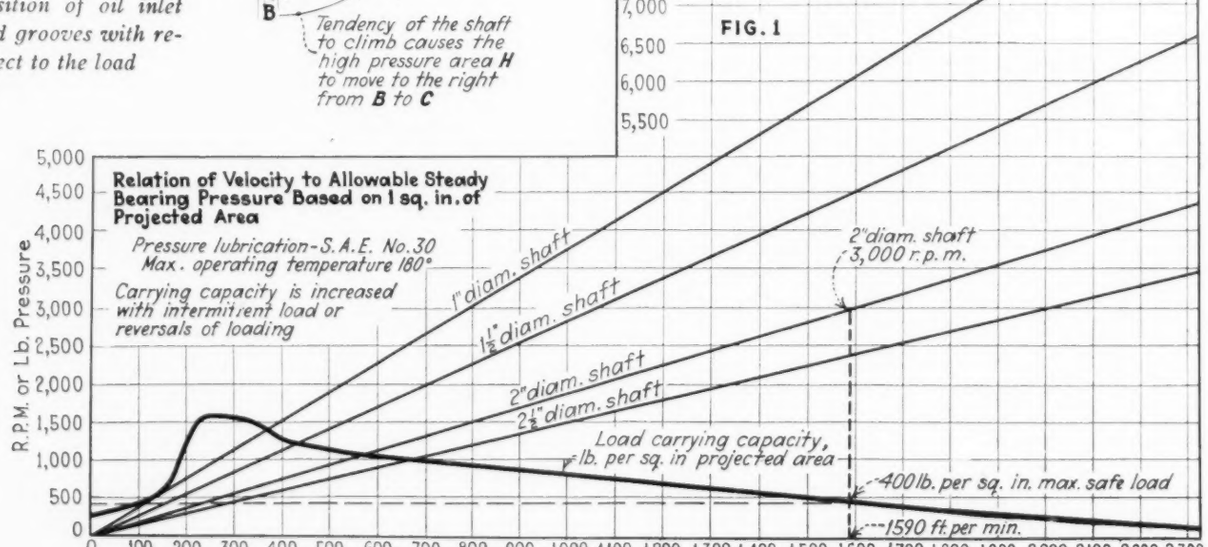


FIG. 1
Relation of Velocity to Allowable Steady Bearing Pressure Based on 1 sq. in. of Projected Area
Pressure lubrication-S.A.E. No.30
Max. operating temperature 180°
Carrying capacity is increased with intermittent load or reversals of loading
1" diam. shaft
1½" diam. shaft
2" diam. shaft
2½" diam. shaft
2" diam. shaft 3,000 r.p.m.
Load carrying capacity, lb. per sq. in projected area
400 lb. per sq. in. max. safe load
1590 ft. per min.
R.P.M. or Lb. Pressure
Ft. Per Min. Surface Speed
0 100 200 300 400 500 600 700 800 900 1,000 1,100 1,200 1,300 1,400 1,500 1,600 1,700 1,800 1,900 2,000 2,100 2,200 2,300
0 500 1,000 1,500 2,000 2,500 3,000 3,500 4,000 4,500 5,000 5,500 6,000 6,500 7,000

Ordering Bearings—I

A proposed form for use when making inquiries of sleeve-type bearing manufacturers to enable them to intelligently recommend the correct material and design. Each item is explained in this and the final article next month

Table I—Bearing Alloys Arranged in Their Order of Shock Resistance

Bunting Alloy No.	Approximate Composition						Sn: Cu Ratio
	Cu	Sn	Pb	Zn	Ni	P	
170	82	13	2	2	1	0.25	0.183
143	85	15	0	0	0	0.05	0.176
164	87	11	1	0	1	0.25	0.150
156	89	11	0	0	0	0.20	0.123
98	28	10	0	2	0	0.05	0.114
51	86	10	2	2	0	0.05	0.116
96	86	10	3	1	0	0.05	0.115
147	87	8	1	4	0	0.05	0.103
27	80	10	10	0	0	0.05	0.125
162	70	10	20	0	0	0.05	0.143
135	77	8	15	0	0	0.05	0.103
72	83	7	7	3	0	0.05	0.084
124	85	5	9	1	0	0.05	0.059
125	75	5	20	0	0	0.05	0.066
158	70	5	25	0	0	0.05	0.067
122	71	4	25	0	0	0.05	0.056
161	67	3	30	0	0	0.05	0.045

load remains substantially the same at all times and is not varied appreciably by the actions of the machine except when starting or stopping.

(B) INTERMITTENT—Intermittent load is as the name indicates. The maximum load that will be applied should be specified.

(C) SHOCK—By shock load is meant that either the bearing or shaft receives an impact regularly. This demands a material with resistance to deformation. The maximum load and the frequency of the shock load should be specified.

(D) LOAD REVERSALS—The direction of the load on the bearing may be reversed during each cycle or rotation as in crank and wrist pins. The maximum load in each direction should be specified, as well as the frequency of load reversals.

The most recent work on the property of the bearing bronzes to resist shock is that which was completed in 1932 at the Battelle Memorial Institute in a program sponsored by the Copper and Brass Research Association. Quoting from their report:

"Figures show quite definitely that, with allowance for lead content, the alloys show resistance to pounding as a function of the tin-copper ratio. Alloys of the type 88-10-0-2 with no lead, and a tin-copper ratio only slightly below that of 80-10-10 with its ten per cent lead should be slightly more resistant. The results bear this out. Next in resistance to pounding after the 80-10-10 comes 70-10-20. Although this alloy has the highest tin-copper ratio of these, the high lead content offsets the stiffening effect of the tin to a large extent. The three alloys 80-10-10, 83-7-7-3 and 85-5-9-1 with approximately equal lead content, have resistance to pounding in the order of their tin-copper ratios. The order of resistance to pounding is the same for 1 per cent deformation and for 5 per cent deformation. There are no direct relations indicated between resistance to pounding and other physical properties, but the general tendencies are for the resistance to pounding to decrease with the hardness, tensile strength and resistance to compression. The resistance to deformation at elevated temperatures was lower for small amounts but roughly the same for deformations of around 5 per cent."

With this information, together with the results obtained on six different representative alloys, the most commonly used bearing bronzes are listed in the accompanying Table I in the order of resistance to shock.

2—Revolutions Per Minute

Shaft diameter in inches times 3.1416 times the r.p.m. divided by 12 equals rubbing speed velocity in ft. per min. or the distance traveled by a given point on the surface of the journal in one minute.

As there is a direct relation between speed and load, the chart, Fig. 1, was constructed. This gives the maximum steady load at any given speed with a well-lubricated bearing. For load reversals and intermittent loading the carrying capacity will be increased greatly because the oil film is re-established each time the load is reversed or removed. For example, there are connecting rod bearings successfully carrying loads 12 times those indicated in Fig. 2. The direction that the shaft

rotates is important because it is necessary to direct the flow of oil from the point of entry across the rubbing surface as shown in Fig. 2. Therefore the direction of rotation must always be given.

3—Action of Part

There are five different kinds of action, listed as follows:

(A) ROTATION—By rotating or rotary motion is meant that the travel is always in one direction, and with this kind of movement also the oil travel is continuously in one direction.

(B) OSCILLATING—By this is meant a to and fro motion or repeated reversals of rotation. Under this condition of travel the oil swishes back and forth but always at right angles to the bearing axis. In addition, a shock load may be applied by the forces causing the movement.

(C) SLIDING—This is a movement to and fro in the direction parallel to the axis, causing the lubricant to wipe out at the ends of the bearing.

(D) THRUST—By thrust is meant the pressure caused by an axial pressure or movement of the shaft against the end of the bearing. This thrust may be light, heavy, shock or dead weight. The possibility of thrust must be considered. The correct material and projected area to carry the imposed radial load on the bearing axis might be recommended, but this same material would pound out under a heavy end thrust.

(E) ROTATION OF SHAFT OR BEARING—By this is meant whether the shaft rotates or the bearing rotates, a factor will cause the stresses imposed on the rubbing surface of the bearing to be vastly different. When the shaft rotates the pressure area between the shaft and bearing remains fixed with reference to the bearing and the bearing material does not get a cold working around its complete circumference. Where the bearing rotates against a stationary shaft, all points of the bearing are passing through the high pressure area and therefore receive an "ironing-out action" or cold working. A bearing metal to resist this action is needed when the bearing rotates whereas when the journal rotates a material that will support the load is all that is required in this respect.

Table II—Effect of Press Fits on the Bore Diameter of Bronze Bearings

The effect of varying press fits in reducing the inside diameter on bearings having wall thickness from $\frac{1}{16}$ to $\frac{1}{4}$ in. The amount of press as indicated by Nos. 1, 5, 12 and 16 is recommended.

Specimen No.	Wall Thickness	O. D. of Bearing	Diam. of Hole	Amount of Press	I. D. Before Press	I.D. After Press	Bore Reduction
1	1/16	1.253	1.251	0.002	1.125	1.121	0.004
2		1.254		0.003	1.125	1.120	0.005
3		1.255		0.004	1.125	1.121	0.004
4		1.256		0.005	1.125	1.1185	0.0065
5	1/8	1.253	1.251	0.002	1.0005	0.998	0.0025
6		1.254		0.003	1.0005	0.9965	0.004
7		1.255		0.004	1.0005	0.9955	0.005
8		1.256		0.005	1.0005	0.995	0.0055
9	1/8	2.753	2.752	0.001	2.500	2.4985	0.0015
10		2.754		0.002	2.500	2.4975	0.0025
11		2.755		0.003	2.500	2.497	0.003
12		2.756		0.004	2.500	2.497	0.003
13	1/4	2.753	2.752	0.001	2.251	2.248	0.003
14		2.754		0.002	2.249	2.2465	0.0025
15		2.755		0.003	2.249	2.2465	0.0025
16		2.756		0.004	2.249	2.2465	0.0025

Because bronze has a higher coefficient of expansion than steel or cast iron, the general belief has been that elevated temperatures would reduce the bore much in the same way as a press fit does. This however does not hold true at temperatures up to 600 deg. F., which is far beyond the operating temperature to which bronze bearings are subjected. On the contrary, the bearing will expand outward, the amount of expansion being limited by the expansion of the housing.

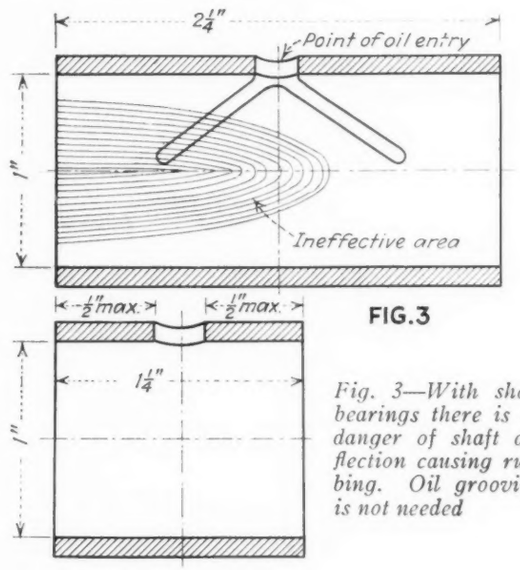


FIG.3

Fig. 3—With short bearings there is no danger of shaft deflection causing rubbing. Oil grooving is not needed

4—Shaft or Journal

Shaft duties are various. Sometimes it is only a question of having the shaft with large enough diameter to carry the torque. In some cases the shaft must carry a bending load or it might have to do both and also withstand shock loads.

There are four factors with respect to the shaft or journal that affect bearing life: (A) Material of the shaft, (B) Shaft deflection, (C) Hardness, (D) Finish.

The first two are a compromise between cost and the shaft size and material needed to hold shaft deflection to a minimum. Shaft deflection cannot be economically controlled by increasing the bearing length as the bearing becomes bell mouthed as indicated in Fig. 3, and fails.

As speeds and loads step up, greater rigidity of the shaft is needed and increased shaft hardness is required, so that the relations of shaft and bearing material stay within workable bounds. Thus, the necessary bronze composition for a definite bearing may demand hardness, consequently a soft shaft would wear rapidly. In addition, the grain structure of a soft steel does not lend itself well to rubbing surfaces and should be used only for low r.p.m. and light loads.

With reference to shaft diameter and finish, to give best results the shaft journal should be hardened and ground, and for high speeds lapped.

5—Housing

Housing is the backing or support of the bearing. This is an extremely important factor as a good bearing can be ruined by a poor support. A good bearing is obtained when engineering and metallurgical data are

correlated to treat the bearing and its housing as one problem.

For example, a bearing to function properly must have intimate contact with the housing around its entire circumference in the sections where support is intended. Because of the conformability of bronze, a proper fit is readily obtained if the bearing is given the correct press fit in the housing. For example, a 1-in. diameter bearing should have a press fit of 0.001 to 0.003 in.

When the bearing material has good conformability, as possessed by cast bronze, it is unnecessary to machine the hole to a close tolerance or finishing with other than a rough boring operation. Not only are savings effected but a better job is obtained. A line ream operation with the bearings in place corrects all errors.

6—Finish in Bearing Bore

Manufacturing procedures that affect the life and operation of bearings may be divided into three parts:

(A) Tolerances allowed and method for finishing the hole preparatory to receiving the bearing.

(B) How is the bearing pressed into hole and how much press fit is allowed; also is bearing locked into housing? When pressing-in, bearings should be located on a pin or arbor and the receiving part held by a fixture.

(C) If burnished, leave just enough stock to clean up, allowing for variations in wall thickness and for the bearings being out of line after they have been pressed into the housing. However, burnishing is not recommended.

If bored or reamed, always leave a minimum of 0.003 in. to be removed from the light side with due regard to out-of-line condition or variance in wall thickness. A cut lighter than 0.003 in. allows the tool to come out from under the skin producing an uneven hole and increasing tool cost. It might be well to add that reaming produces 40 to 50 per cent bearing surface and diamond boring 50 per cent to 80 per cent bearing surface, depending in each case on the accuracy of the machining equipment.

The finish of bearing or rubbing surface is important and has a direct influence on the life and performance of the bearing. In the first place the bore of the bearing must be in line with the shaft and, to prevent oil shear, must have a smooth finish.

As loads and speeds increase harder and smoother shafts and bearings must be used.

7—Clearance Between Shaft and Bearing

Running clearance is governed by the size and speed of the shaft, finish of the bearing and shaft, and operating temperature. Generally speaking, a clearance of 0.001 in per in. shaft diameter is common practice, with a minimum of 0.005 in. on shafts ½ in. diameter or less.

Clearance must be great enough so that when the bearing comes up to operating temperature, it will not force a metal to metal contact. Yet the bearing clearance must be small enough to prevent shaft chatter.

Inquiry Form for Sleeve-Type Bearings

1. Total Radial Bearing Load——lb
Type of load:
(a) Constant.....——
(b) Intermittent...——
(c) Shock.....——
(d) Load reversals.——

2. Speed——r.p.m.
Direction of rotation when facing the end of the bearing;
(a) Clockwise.....——
(b) Counterclockwise——

3. Action of Part
(a) Continuous rotation——
(b) Oscillation.....——
Angle of oscillation——deg.
(c) Sliding...——
(d) Thrust load——lb.

4. Shaft or Journal
(a) Material specifications——
(b) Diameter——in.
(c) Length (max. permissible) ——in.
(d) Hardness, ——Brinnell——
(e) Finish, Machined ——
Lapped ——
Ground ——
Polished ——

5. Housing
(If possible, submit detail drawing giving controlling dimensions and tolerances)
(a) Material——
(b) Finish of housing bore:
Machine bored——
Reamed.....——
Broached.....——
(c) How is bearing to be pressed into housing.
Arbor press——
Will mandrel be used?
Will bearing be piloted into hole?——

6. Finish in Bearing Bore
Diamond bored——Reamed
Broached——Burnished——

7. Clearance Between Shaft and Bearing
Required by design,——in.
(Bearing manufacturer will recommend clearance, if so requested)

8. Character of Lubrication
(a) Oil—— (b) Grease——
(c) Continuous lubrication assured by lubrication system
(d) Lubricant furnished periodically——
(e) Type of lubricating fitting
(f) Grooved bearing desired——
Type of grooving——

9. Operating Conditions
(a) Heat, max. temperature ——deg. F.
(b) Cold, min. temperature ——deg. F.
(c) Dirt and dust, (described)
(d) Acids or corrosive liquids or fumes,——
(e) Would leakage of lubricant damage the products of the machine or be objectionable otherwise?——

10. Space Available for:
Bearing length——in. (*See item* 4)

11. Space Available for:
Outside diameter of bearing ——in. (*See item* 4)

12. Unusual Service Conditions——

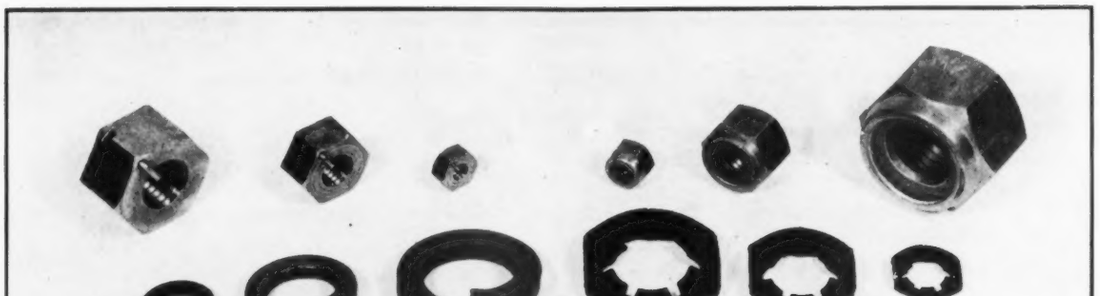


Fig. 1—Various locking devices which assure the safety of assembled parts

Safety Fastenings

An analysis of the locking action and the application of miscellaneous screw and nut locks

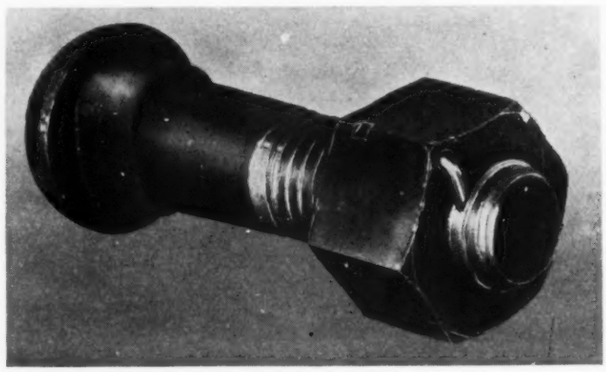
BECAUSE the failure of even a minor part of a machine results in a waste of time and effort for repairs, and because such mishaps may result in personal injury or even death, much attention is given to the small details that control the safeguarding of mechanisms. Among these are safety fastenings for screws and nuts, of which there are a great variety.

One of the simplest locking methods is to merely apply additional axial pressure between the bolt and nut thread. This takes the form of a plain spring washer, which, when compressed, exerts longitudinal pressure only. In many cases this does not assure sufficient locking action and the purpose of the numerous types of patented locking devices is to overcome this deficiency and at the same time not interfere unduly with disassembling the nut.

To hold a nut more positively in a position relative to the face against which it bears, a spiral spring washer having two opposing barbs, an example of which is made by The Positive Lock Washer Company, and shown at the left center in Fig. 1. These barbed ends, which are slightly tapered, deflect momentarily when compressed to a flat position, reacting so as to force the barbs into the nut and the surface against which it bears.

Whereas this barbed washer connects the nut and the clamped member, another type of locking device connects the nut and the bolt. This can be accomplished by a pin lock incorporated in the nut, such as the type shown in the upper left of Fig. 1, which is manufactured by the Automatic Nut Company, Inc. A notched alloy steel pin is assembled radially in one corner of the nut and it is held in place by swaging a portion of this corner into the pin notch. The end of the pin, which is shaped to the thread form, travels between the threads and assumes a fixed biting angle. Any turning of the nut tends to make the pin bite into the bolt at the root of the thread. To remove the nut,

Fig. 2—When assembled, torsion makes pin bite into root of thread



a wrench is needed to force the pin to the side opposite from the biting position. It is claimed that no damage to the pin, thread, nut or bolt results from repeated removal. Fig. 2 shows the pin locking a nut on a bolt.

Pressed Steel Lock

Embracing the root of the bolt thread at numerous points on the circumference, and screwed on top of a hex nut, is the method used by The Palnut Company (center right, Fig. 1). Made of thin flat stock, this lock nut is formed with turned up edges into the shape of a hexagon having an internal serrated cone. These serrations produce a number of prongs which follow the helix angle of the bolt thread. After the device contacts the hex nut, an additional $\frac{1}{4}$ to $\frac{1}{3}$ turn locks it by flattening the prongs, causing them to bind at the apex of the root of the thread. All stud and bolt nuts on the Wright Cyclone airplane engine, Fig. 3, are locked by this device.

When a nut is tightened, the sides of the thread nearest the clamped faces come into contact with one another. This principle is used in the nut manufactured by the Elastic Stop Nut Corporation by inserting a fiber collar as shown at the upper right in Fig. 1. The inside diameter of the fiber ring is equivalent to the tops of the nut thread. The bolt thread is said to be impressed into the fiber collar, but will not cut it. The collar then acts as a brake against vibration, and incidentally also seals the nut against corrosion. This type of nut is good for temperatures up to 250 deg.F. Fig. 4, left, shows the bolt starting to enter the fiber ring, and at the right is shown how the fiber collar hugs both sides of the thread when assembled. In Fig. 5 is shown how the spring clip is fastened to the tubular axle in the front end suspension of a Hupmobile car, using four of these lock nuts. They must not only serve to prevent the nuts from loosening, but also keep water from reaching the bolt threads.

By bringing the elastic characteristics of the metal in the bolt and nut into play, locking action may be obtained by engaging corresponding tapers at the root of the bolt thread and at the crest of the nut thread. A thread designed to accomplish this is the Dardelet self-locking screw thread, shown at the lower right, Fig. 1, and in Fig. 6. The sides of the thread

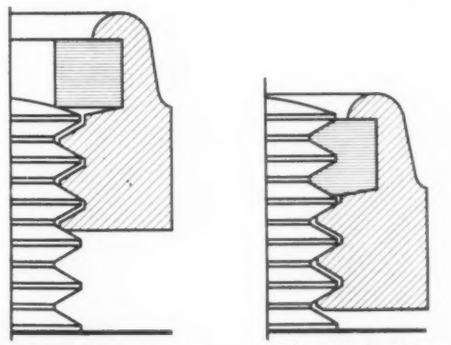
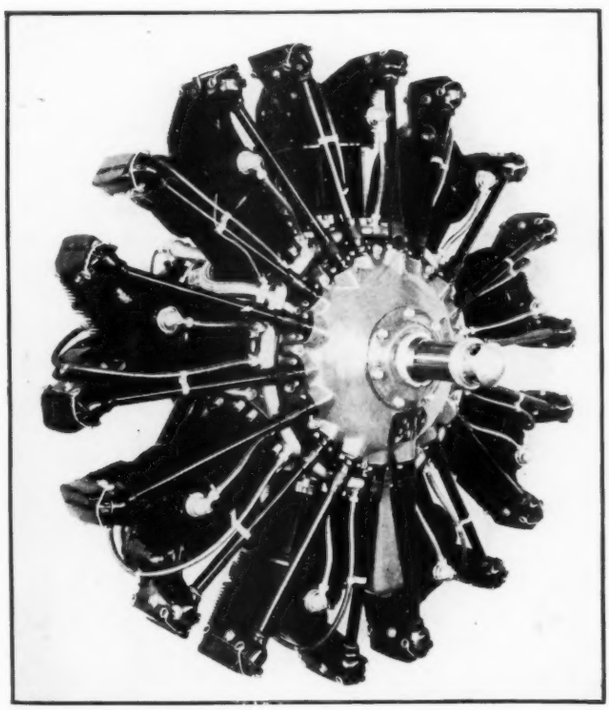


Fig. 4—The bolt thread is impressed into the fiber collar, resulting in a gripping action which retains the nut in a fixed position

Fig. 3—Personal safety is a requirement in the Wright Cyclone engine. Stud and bolt nuts are secured with the device shown at the right



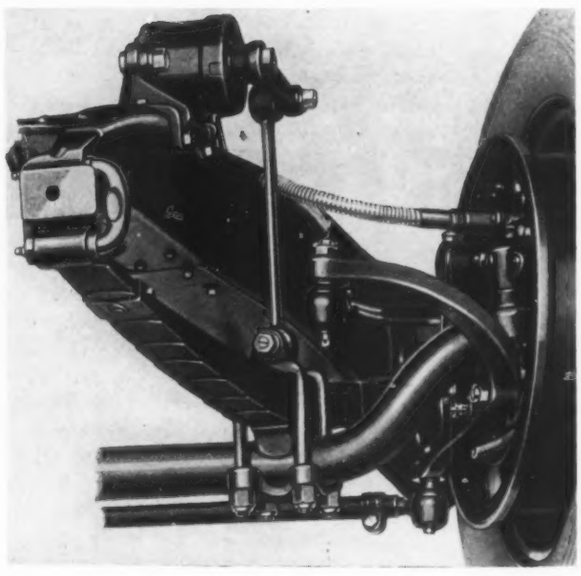


Fig. 5—Front end suspension on Hupmobile, with four stop nuts to clamp the spring clip below tubular axle

are formed similar to Acme thread, with a wide conically tapered surface at the root of the bolt thread, and a narrow conically tapered surface at the crest of the nut thread. The nut is free running until it meets resistance, when the angular surfaces engage and lock.

Special Threads

The abutting sides of the thread carry the major part of the tensile load. The threads are more shallow than standard threads, giving greater strength at the root diameter of the bolt. In one instance, it is used for a self-locking nut which clamps a ball bearing on a shaft as shown in Fig. 6, eliminating the use of a lock washer and shaft keyway.

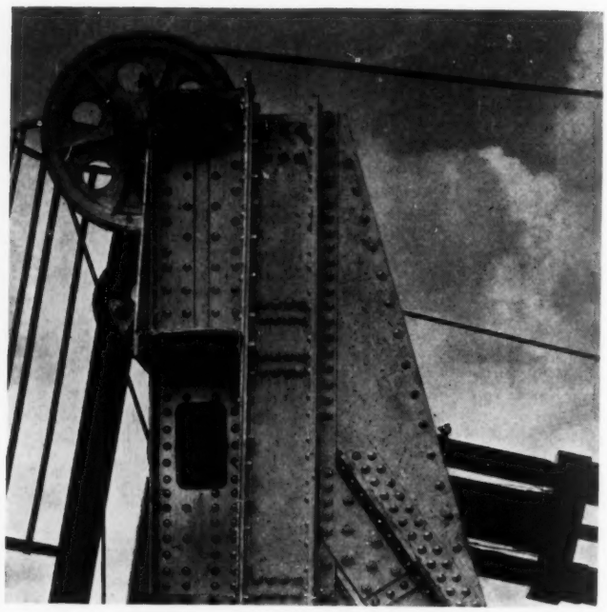


Fig. 7—Sheave mounting for the upper portion of a head tower on 25-ton cableway, using rivet bolts

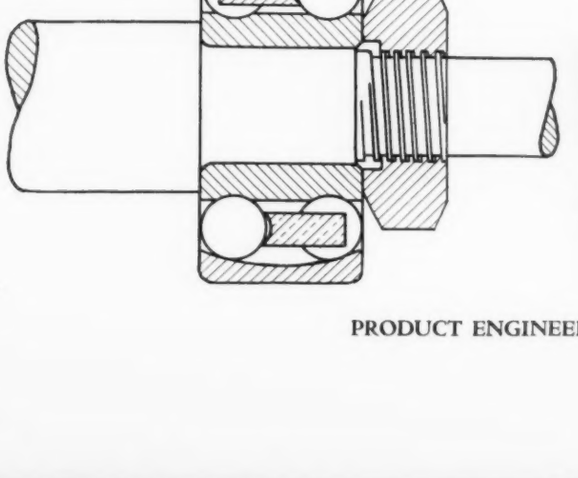


Fig. 6—With contacting tapers at the bottoms of the threads, a locking action is obtained through the high pressure developed at the metal to metal contact

In the construction machinery shown in Fig. 7, Dardelet "Rivet-Bolts" are used with hex nuts. These bolts have a fluted shank or grip portion, the outside diameter of which is slightly greater than the standard drilled or punched hole. When driven into the hole, the tops of the serrations are deformed and are said to give a tight body-bound fit. The 6 deg. tapered faces of bolt and nut threads provide the gripping and locking action. The material used for the bolts has 70,000 lb. per. sq.in. tensile strength, being a carbon manganese alloy possessing also high shear value.

When enough pressure can be brought to bear on all surfaces of the thread, so that little or no clearance remains, a thread lock is obtained. Parker-Kalon use this method on their hardened drive and self-tapping screws which make their own thread grooves when driven into a hole having a diameter of approximately the root diameter of the screw. The displaced metal is pushed against the screw body at the bottom of the thread, closing in on all surfaces for retention.

Drive screws are used for making fastenings to sheet steel, cast iron, phenolic materials, slate, ebony and asbestos. For heavier materials these screws with hex heads are used. Screws that are driven in with a screw-driver or wrench have a thread similar to a wood screw, except that this thread is spaced wider apart. Drive screws have a long helix angle of thread and rotate slightly as they are hammered in.

The typewriter portrayed in Fig. 8 is assembled with self-tapping screws. The frame is made of 0.050 in. sheet steel and clearance holes are provided in the fastened parts. To test the security of 68 points of assembly, a vibration test at 565 vibrations per minute failed to show after $5\frac{1}{2}$ hours any evidence of looseness in these fastenings.

In Fig. 9 the center bracket is fastened to the bar assembly of the typewriter by self-tapping screws. This assembly is made entirely of sheet metal.

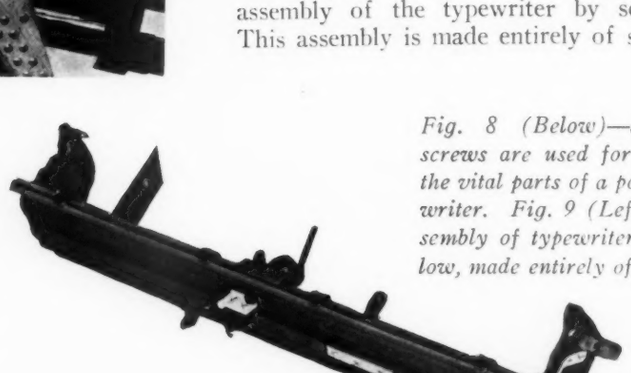
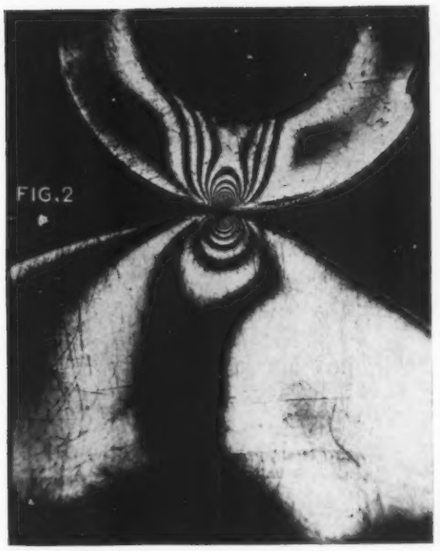


Fig. 8 (Below)—Self-tapping screws are used for assembling the vital parts of a portable typewriter. Fig. 9 (Left)—Bar assembly of typewriter shown below, made entirely of sheet metal

Analysis of Stresses in Cam Rolls



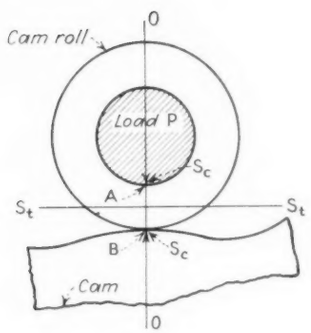


Fig. 1—Diagram of forces acting on a cam roller

W. J. HART
Engineering Division, Sharon Works
Westinghouse Electric & Manufacturing Company

COMPARATIVELY little data have been published concerning stresses in hollow cam rolls, or permissible load stresses for certain materials in such mechanisms. Most cam rolls are designed with but limited knowledge of the stresses involved.

The most common type of cam roll operates on a pin through its center and fortunately this type lends itself well to stress analysis. Fig. 1 suggests that the region between the pin and the cam is under compressive stresses of varying magnitude.

Differences of stress magnitude are apparent from Fig. 3(*a*) which assumes a parabolic distribution of stress at the inside surface of the hole. On this assumption it is clear that in transferring the load *P* to the cam the unit stress on the inside surface of the hole is smaller than the unit stress at the periphery of the cam and roll. It follows then that a critical stress occurs at the contact surface of the cam and roll which critical stress is known as the Herz contact stress. Since the rectangular area of contact between the cam and roll is quite small, the resulting contact stress at this point is high for a small applied load *P*. Consequently when one calculates the load *P* applied against the contact surface in terms of lb. per sq.in., the resulting stress is far beyond the established limits for the elasticity of steels; it has been known to exceed 700,000 lb. per sq.in. and yet without signs of failure.

Unfortunately we cannot see what takes place inside the roll and we cannot in that manner determine what displacements of cube lattices may have occurred. However we can analyze the condition by passing polarized light beams through an exact model of the cam and roll made from bakelite. When subjecting such a model to a definite compressive force, stress differences will be detected in the color fringes (isochromatic bands).

Photoelastic methods do not indicate principal stresses but only their differences. For this reason, an examination of a photograph will not reveal the actual value of the compressive or tensile stresses. Methods for obtaining these principal stresses when their differences are known, have been developed, but these methods either are not entirely reliable or they involve undue labor.

In making the photoelastic tests, a bakelite model was put in a testing frame and a load was applied to the cam and roll by means of weights. A picture of this model, shown in Fig. 2, was taken with polarized light while it was under stress in the test frame. The picture shows marked evidences of the transfer of the stress boundary of the cam roll pin. The isochromatic bands indicate that the pressure distribution on the inside of the cam roll is parabolic as suggested in Fig. 3(*a*). It is assumed that the resultant compressive force is accompanied by the natural tensile component perpendicular to it.

To understand the fact that a cam roll does not fail under high computed compressive stresses, the transfer

of its compressive load to a tensile component must be taken into account. To analyze a simple condition, let us take an actual load of 45 lb. on the bakelite model having a diameter of $1\frac{1}{2}$ in., a width of $\frac{1}{4}$ in. and a $\frac{3}{4}$ in. diameter hole. By experiment on a bent specimen of this bakelite it was found that each fringe corresponded to a difference in principal stresses equal to 266 lb. per sq.in.

To estimate the maximum tensile stresses in the roll, let S_c represent the principal stress (compression) in the direction of, and along, the line OO, Fig. 1. Let S_t represent the principal stress perpendicular to the line OO. The principal stress difference $S_t - S_c$ is determined by counting the number of fringes of zero order. S_c is taken with a negative sign since the stress is compression.

Near the point corresponding to A of Fig. 1, there are at 45 lb. about $3\frac{1}{2}$ fringes, which indicate a stress difference of

$$S_t - S_c = 31/2 \times 266 = 931 \text{ lb. per sq.in.} \qquad (1)$$

To estimate the tension stress S_t, an estimate of S_c must be obtained. From Fig. 3(a) it will be seen that the contact pressure extends over an area about $\frac{1}{2}$ in. wide. Assuming that this pressure distribution is parabolic, the average value of compression stress with a 45 lb. load will be $\frac{45}{\frac{1}{2} \times \frac{1}{4}} = 360$ lb. per sq.in. The maximum value with a parabolic distribution is 1.5 times the average value, or in this case $1.5 \times 360 = 540$ lb. per sq.in. Taking $S_c = 540$ lb. per sq.in., the indicated value of tension stress from Equation (1) is $S_t = 931 - 540 = 391$ lb. per sq.in. This estimate will hold for the region surrounding point A in Fig. 1.

To estimate the tensile stress near the point of contact B, the results obtained by Goodier (A.S.M.E. *AM Vol. 54 No. 18*) may be used. He considered tension stresses, perpendicular to the line of action of the forces, in blocks loaded as in Fig. 3(b). An investigation of blocks having three different ratios of width to depth, indicated that the maximum tension was equal to:

$$S_t = \frac{3.64\,P}{\pi\, bt} \qquad (2)$$

where P = load, b = width of block, t = thickness.

It seems reasonable to assume that Equation (2) will give an indication of the maximum tension stress in the neighborhood of the contact point between cam and roll. Using $P = 45$ lb., $b = \frac{3}{8}$ in., $t = \frac{1}{4}$ in., the tension stress near point B in Fig. 1 becomes:

$$S_t = \frac{3.64\,P}{\pi\, bt} = \frac{3.64 \times 45}{\pi \times 3/8 \times 1/4} = 557 \text{ lb. per sq.in.}$$

This calculated tension stress is probably on the safe side; it is very likely that the actual tension is less than that computed because of the bending action of the parabolic distribution of load on the inside edge of the roll. Comparing this value with 390 lb. per sq.in. found for locations near point A in Fig. 1, it is seen that the two stresses are not of a widely different order of magnitude. It appears therefore that a conservative estimate of the tensile stresses in the roll may be had by using Equation (2), taking t equal to the width of the roll and b equal to its thickness.

It is interesting to make comparisons between the results of ordinary calculations and those based upon the method of analysis which has been described. For

Table I--Application Factors For Various Materials and Services

Material	Application	Load	Service	Hardness	β	S_e
Cold Rolled Steel	Static	Light	Cont.	20 Scl.	7	38,000
	Dynamic	Light	Intermit.	20 Scl.	7	
Axle Steel	Static	Light	Cont.	30 Scl.	5	45,500
	Dynamic	Light	Intermit.	30 Scl.	5	
Tool Steel	Static	Heavy	Cont.	60 Scl.	4	87,000
	Dynamic	Light	Cont.	60 Scl.	4	
	Static	Light	Cont.	60 Scl.	7	
	Dynamic	Heavy	Cont.	60 Scl.	7	

Fig. 3—At (a) is shown the parabolic pressure distribution as indicated by the isochromatic bands in Fig. 2. At (b) is shown the fundamental force diagram

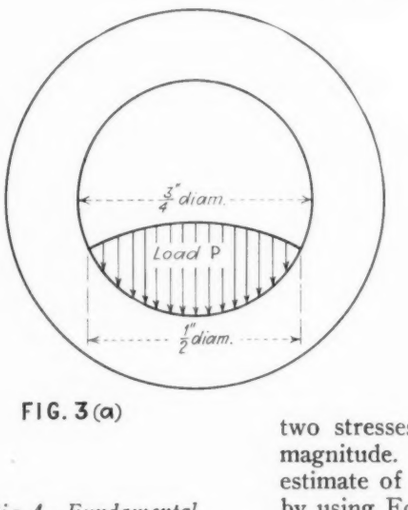


FIG. 3 (a)

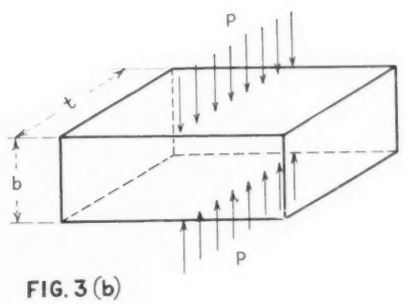


FIG. 3 (b)

FIG. 4

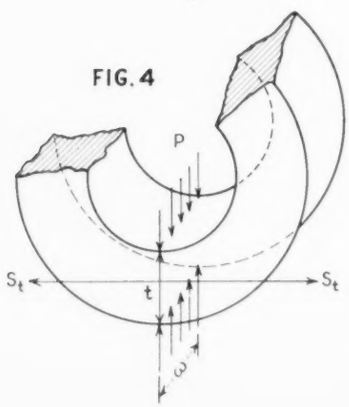


Fig. 4—Fundamental force component diagram for a cam roll of thickness t *and face width* w

this purpose, assume a steel roll having the same dimensions as the bakelite model except for a width of $\frac{1}{2}$ in., subjected to a load of 2,500 lb. The Herz contact stress, computed by the usual methods, would amount to approximately 300,000 lb. per sq.in., a value far in excess of a practical working stress.

On the other hand, calculations based upon the method of analysis which has been described, indicate that a load of 2,500 lb. on the roll will produce a tensile stress of 14,260 lb. per sq.in. Actual tests with a load of 2,500 lb. disclosed no signs of failure.

Equation (2) will hold for rolls having a pin diameter one-half the roll diameter and a cross section of metal that is nearly square. A more general equation is as follows:

$$P = \frac{S_e \pi A}{(\beta + \alpha)\, 3.64} \qquad (3)$$

Where

P = Maximum roll load in lb.
S_e = Endurance limit of material in lb. per sq.in.
A = Cross sectional area of roll ring in sq.in. $(w \times t)$
β = Material constant and application factor (Table I)
$\alpha = \left(\frac{d_1 + d_2}{d_1 \times d_2}\right)$ (See Fig. 5)

In designing a cam and roll we are concerned with four major dimensions, namely: (1) diameter of the cam; (2) diameter of the roll; (3) diameter of the roll pin; (4) width of the cam roll.

It is recommended that the use of Equation (3) be limited to cam rolls ranging from $\frac{1}{2}$ to 2 in. in diameter and operating in conjunction with cams not exceeding 8 in. in diameter. In view of this consideration, the graph of Figure 5 has been prepared to assist in the calculation of rolls within this range.

Equation (2) is useful for calculating the proper dimensions of a cam roll for a given load. However, other factors enter into the design of a roll and to a large extent determine its success or failure. Let us, for example, consider the load imposed upon a cam roll for the following various conditions: (1) intermittent light static load; (2) intermittent heavy static load; (3) intermittent light dynamic load; (4) intermittent heavy dynamic load; (5) continuous light static load; (6) continuous heavy static load; (7) continuous light dynamic loading; and (8) continuous heavy dynamic loading.

In applying the formula, Equation (2) to a specific problem, it is necessary to consider all the various factors that affect the roll. Table I shows a few of the more common materials and applications and their respective constants.

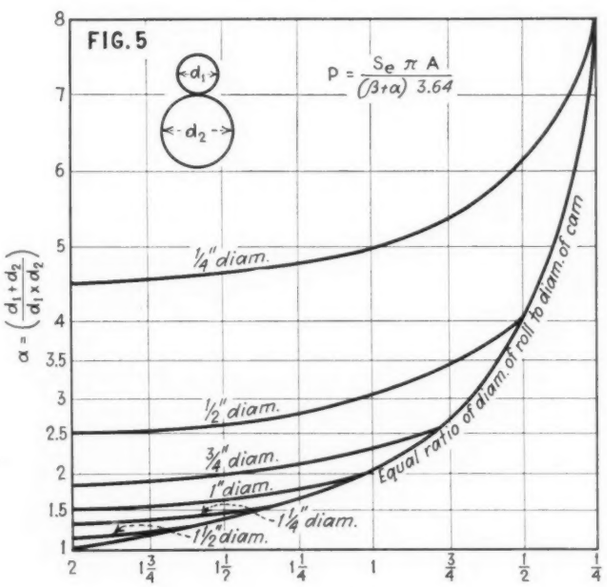


Fig. 5—Curves for obtaining the value of α for various ratios of roll diameter

Lubrication and Manufacture

The consideration of lubrication is sometimes of major importance in calculating cam rolls, especially when the service involves continuous heavy loads. In such cases, the unit load upon the bearing surface between roll and pin must be kept within the safe rupturing limit of the lubricant. If this is not taken into consideration, excessive heating and wear are encountered, tending to shorten the life of the roll.

The machining of cam rolls is of vital importance, especially when the roll is to be heat-treated. Both inside and outside surfaces, and also the sides, should receive a mirror polish to eliminate tool marks for quite frequently it is from this source that hardening cracks originate. Also, generous fillet radii are recommended whenever possible to prevent flaking.

The selection of a suitable material is often a problem. The allotropic modifications of iron in general give us a wide selection of grain structures and hardness, and since the carbon content has a direct bearing upon the hardness of the material, it should be given special consideration. The higher the carbon content, the greater the risk of not obtaining a satisfactory roll because of the dangers of getting residual stresses and quenching cracks in the heat-treatment.

Design of Cam Rolls

Whenever possible the designer should use rolls as large as possible and with tensile and contact stresses as low as possible so that it will not be necessary to heat-treat. A great many rolls are made from expensive tool steels for applications that require only ordinary machine steels. The rolls should be designed with the proper ratio as given in Fig. 5 and the constants given in Table I, and side drafts should be provided to reduce friction in the roll forks.

The tensile-stress method of computation is dependable for calculating cam rolls and the resulting stresses may be compared directly with the known elastic constants of all steels. In comparing permissible tensile stresses of materials, it is recommended that the endurance value of the material be used instead of the static test value, for as it is well known that with repeated loading and unloading, or with reversed stresses, failure can be produced by stresses that are smaller than the ultimate strength of the material as obtained from a static test.

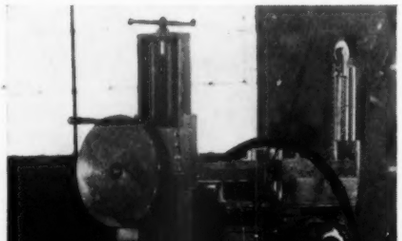
QUESTION AND COMMENT

Sawing With An Electric Arc

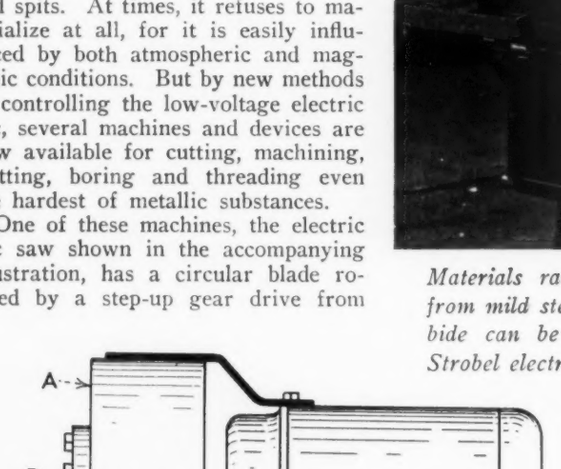
M. C. SMITH
Rich Manufacturing Company

● The free low-voltage electric arc is temperamental and elusive. When struck from the tip of a welding electrode, the arc often sputters, spins and spits. At times, it refuses to materialize at all, for it is easily influenced by both atmospheric and magnetic conditions. But by new methods of controlling the low-voltage electric arc, several machines and devices are now available for cutting, machining, slotting, boring and threading even the hardest of metallic substances.

One of these machines, the electric arc saw shown in the accompanying illustration, has a circular blade rotated by a step-up gear drive from



Materials ranging in hardness from mild steel to tungsten carbide can be machined on the Strobel electric arc saw



By conducting the arc current around the helix C *surrounding the blade* D, *the blade is magnetized, controlling the arc*

an electric motor. As shown in the drawing, the blade *D* is surrounded by an electric coil or helix *C* so that the center of the saw blade becomes one pole of an electromagnet and the periphery of the blade becomes the opposite pole.

The arc-generating electric current enters the unit through the non-magnetic housing *A* and is fed to the coil or helix, and from the coil is conducted into the blade by the brush *B* and permitted to arc to the work. In this construction, the arc is controlled by an electromagnet energized by the current that forms the arc.

The blade is toothed, not because the blade must actually cut, but to produce voltage oscillations which assist in directing the arc as well as giving the arc its velocity. The lengthening and shortening of the low-voltage arc, as the teeth of the blade pass the work, produces a variation in the voltage, setting up oscillations of about 2,000,000 a minute. It is this rapid oscillation of current that directs the electric arc from the tips of the saw teeth to the work so that the arc path is not more than 0.005 in. wider than the blade.

During the cutting operation, the metal in the kerf is fused and swept out so rapidly that no section of the material being cut is exposed to the excessive temperature long enough to induce an appreciable change in the structure. Likewise, the rapid rotation and the short interval during which the arc is passed from a given tooth does not allow the saw blade to become heated.

Although the voltage never becomes much above 20 volts, the current may reach 1,000 amp. or more, depending upon the thickness and melting point of the metal being cut. Sections of mild steel may not require more than 100 amp., but harder materials will need much higher currents.

Better Adjustments

F. K. LAWLER
Jackson Heights, N. Y.

● Machine designers are pretty good at turning out a product with good inherent performance and wearing qualities. But these same designers often overlook the fact that satisfactory long-time service depends as much upon the machine remaining in proper adjustment as it does upon good design, materials and workmanship.

Consider the automotive designer. He turns out a motor inherent in the design and construction of which is, say, 80 hp. of performance and 50,000 or more, miles of wear. But how many of his motors are developing 80 hp. after 10,000 miles? Possibly one in a dozen. Possibly less. Why? For no better reason than that some minor adjustment is not what it ought to be. Perhaps the timer contact points are not breaking wide enough. Perhaps the carburetor setting is a quarter turn too rich. Perhaps the valve clearance is 0.010 in. when it ought to be 0.006 in. Maybe the ignition timing is 4 or 5 deg. too slow. Most industrial machines, too, have parts which must be adjusted within close limits to get the highest degree of performance.

Designers may reason that it is the service man's job to keep machinery in "tune." But that argument won't hold water. Only a fraction of those workmen calling themselves mechanics are really capable of adjusting a machine so as to get out of it the performance which the designer puts into it. And, as a matter of fact, the mechanic can hardly be expected to know what is called for by the blueprints in the designer's office.

The machine manufacturer ought to see to it that as many as possible of the adjustments are made "permanent" and that specifications for the others are permanently and prominently displayed near the point where the adjustment is to be made. Some automobile engines, for example, carry the limits of valve clearance on the cover which must be removed to make the adjustment. Moreover, the means of making adjustments should be made as nearly foolproof as possible. The method of timing the antedated model A Ford is an example of this.

And in the matter of factors vital to good performance, there usually are two or three parts which, while they never need adjustment, will seriously affect the operation of the machine if they deteriorate or wear slightly. To revert to the automobile, there are the condenser and the coil. Here again, the designer knows the vital parts and he should put into those parts the utmost in quality. Especially is this true when, as in the case of the coil and condenser, the cost is practically nothing compared with that of the entire machine.

If designers of industrial machinery think they do better than the automotive engineer, they have but to spend a little time in factories using filling, closing and wrapping machinery to change their minds. Haywire adjustments on these machines are a more or less constant source of production delay. In a chewing gum factory, to cite a specific case, the mechanics spend so much time keeping the wrapping machines going that they consider the regulation and maintenance of the photoelectric register controls nothing at all.

Practical Professors

To the Editor:

● Engineering college courses should offer continued instruction in and emphasis on the bases upon which engineering knowledge is founded, analytical methods of attack, oral and written self-expression, the use of reference sources, and economics of design.

Average college preparatory work fails to implant or emphasize the idea of thinking in basic terms. Such analysis or analytical ability as may be partially developed in high school is largely the result of the smattering of algebra and geometry to which the student has been exposed. Preparatory training usually sacrifices brevity and exactness of expression to more rhetorical and florid style, and the use of reference works is only partially developed. Finally, ideas of cost per unit result, though capable of elementary illustration, are only barely mentioned.

When such deficiencies exist, college training is consequently slow. Usually, the first year is spent trying to develop analytical faculties and to kill off mental laziness, endeavoring at the same time to correct errors in English, and to teach new vocabularies. Other departments are also exposing the student to some modern language, imparting some fundamental notions on chemistry and physics, and bringing algebra, geometry and perhaps trigonometry up to a state of incipient utility.

Economics of design is given good consideration in numbers of courses in the higher grade institutions, the treatment often being better in elective than in required series of lectures. Any really good course in materials includes many of the items mentioned by H. E. Taylor in the October number, page 388. The additional requirement of one to one and one-half years' commercial (not college) shop practice over and above the academic requirements would help the student formulate intelligent questions for discussion.

Teaching definitely includes the exercise of ability to patiently impart inspiration, knowledge, ideas and discrimination. Without this ability the best technologist remains a mediocre teacher. Even with it, successful teaching of engineering courses is hard work in the face of the lack of preparation and the mental habits of the usual high school graduate.

Recognizing these conditions, and realizing that the profession is not too highly paid, one wonders where the "practical professors" are to be obtained in quantities sufficient to serve all the students now enrolled in engineering colleges.

—Roland V. Hutchinson
Lansing, Mich.

Retaining Devices

To the Editor:

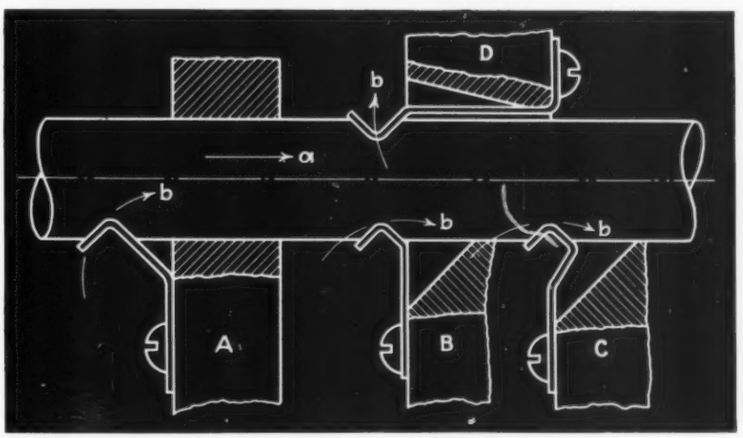
● Snap springs such as illustrated in Fig. 64 on page 391 of the October number of *Product Engineering,* will jam against the sliding rod when the rod is moved in the direction shown by the arrow *a* in the accompanying diagram. Even if the spring has the form and arrangement shown at *B* the same jamming tendency exists and is only eliminated by the symmetrical disposition of the retaining hump as at *C*. At *D* is shown the only safe arrangement of this simple type of spring retainer, and even here the simplicity over the seemingly more complex spring and plunger type is more apparent than real, for extra work is entailed in the grooving operation.

—Francis W. Shaw
Broadley, England

To the Editor:

● In reply to Mr. Shaw's discussion, the design shown in Fig. 64 in the October number was an actual and satisfactory installation used for a belt shifting rod. The spring shown in this design was made of steel, the hump being bent to an included angle of 90 deg.

If the coefficient of friction is smaller than the tangent of half the angle of bend, such an arrangement will be successful. The design submitted by Mr. Shaw, as shown as *B* and *C* would give too much bending angle to the detent. The method shown at *D* has merit, and the objection of cutting the keyway in the hole to clear the spring could be



Retaining spring design suggested by Mr. Shaw, shown at right, as compared with design at A, published in October

met by milling the shaft. The spring would then project into the round hole without additional machining. In this instance, the screw should be on the outside of the case to facilitate assembly. —ADAM FREDERICKS

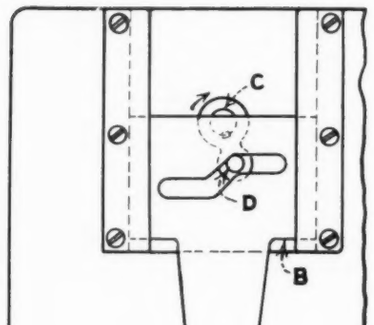
Rectilinear to Rotary Motion

J. E. FENNO
Belleville, N. J.

● In an automatic candy wrapping machine, it was necessary to drive a rotating shaft from an oscillating lever. To carry the drive past dead center the cam mechanism shown in the accompanying figure was developed.

The oscillating lever *A*, pivoted to the machine frame, is connected to the crosshead *B* which slides in stationary guides. The cam slot in the crosshead rotates the shaft *C* by means of the crankpin *D*, keyed to the shaft.

In the position shown, the lever is at the lowest point in its travel, but the crank has not yet passed its lower dead center position. However, as the lever moves upward, the angular part of the cam slot swings the pin toward the left and past the dead center. The horizontal portion of the slot at the left then continues the rotary movement.



Inclined slot in the cam allows the oscillating lever to drive the crank past dead center

At the upper end of the stroke the crank is carried past the top dead center in the same manner by the upper face of the cam slot. To provide a smooth action between the pin and cam slot at the point of reversal, the junction of the inclined groove and the horizontal slot is rounded.

Cam Contours

To the Editor:

● Exception can be taken to two ideas in the article "Cams for High Speed" by C. N. Neklutin appearing in *Product Engineering* in the July number. The statement, "shocks caused by a sudden change in acceleration" made on page 250, is surprising as the great majority of engineers undoubtedly consider a shock as caused by the rapid change in the velocity of a body. In most high-speeed machines the amount of acceleration is much more important than its rate of change, since the maximum value of the inertia force determines the stresses and reactions. Any change which involves an increase in the maximum value of the acceleration may well be a step in the wrong direction.

On the same page, Mr. Neklutin states, "Undoubtedly the most commonly used acceleration and deceleration curve is the sine curve . . ." but the sine curve shown in the article is not the common harmonic motion, as the 360 deg. of motion shown is for only a single stroke instead of a complete revolution of the cam. The most common curve for slow speed cams is the constant velocity type, while for high speed machines harmonic motion, parabolic or constant acceleration, and combinations of arcs of circles are most common for approximating constant acceleration. I have never seen the sine curve as proposed by Mr. Neklutin used on any machine.

The author refers to the "Scotch Yoke" and eccentric cams as having a point of shock at each end of the stroke. While these are the points where the acceleration is a maximum, there is no shock in the sense in which he uses the term, because the motion is continuous and the acceleration does not change in direction. There would be a sudden change in acceleration only if the motion started from a dwell.

In the design of high speed machinery it is desirable to keep acceleration as low as possible without giving particular attention to any easeoff in the reversal of direction or sudden change in the amount of acceleration. If under some conditions an easeoff seems desirable, an arrangement in which the acceleration varied as a constant multiplied by the time would be much simpler mathematically than the sine curve. —JAMES A. HALL
Brown University

To the Editor:

● With reference to C. N. Neklutin's articles on "Cams for High Speeds" it appears that some fundamental relations have been overlooked. The author evidently has in mind a cam with a dwell, the follower starting from rest and ending there. However, neither the acceleration-time curve in his first illustration, nor the speed-time curve in the second one, represent such a period. The action of the cam would have to start at the crest or trough of the sine curve representing acceleration, and halfway up at zero speed on the curve representing speed. Furthermore, for any type of complete cyclical motion, where the follower returns to its original position, it is necessary that the area included in the speed-time curve above the horizontal axis be equal to that included under the curve.

The author also states that by using constant acceleration, cam shock will result. There may be some question as to the definition of the word shock. An object hanging by a string and allowed to drop when the string is cut, could hardly be said to start with a shock. Yet, it would have constant acceleration. An abrupt change in the direction of the travel-time curve means a shock or large acceleration limited only by the elasticity of the materials, but this is not the case for the speed-time and acceleration-time curve.

Curves for both simple harmonic motion and constant acceleration as applied to reciprocating motion are given in the accompanying diagram. Comparing the speeds and accelerations the constant acceleration motion gives the lower value of maximum acceleration to the follower and it also imparts a higher speed than does simple harmonic motion.

Where R is one-half the total travel and T is the time for one cycle, the maximum velocity V and the maximum acceleration A for simple harmonic motion is approximately,

$$V = 6.28\,R/T$$
$$A = 39.6\,R/T^2$$

For constant acceleration these values become,

$$V = 8\,R/T$$
$$A = 32\,R/T^2$$

Although the two travel-time curves are similar in appearance, they give widely different speed and acceleration curves. —ROBERT PASH
Boesch Manufacturing Company

To the Editor:

● In the fourth paragraph of the first article, the problem is limited to the analysis of a cam stroke, "as being the movement of the cam follower from one position of rest to the next position of rest." Possibly this definition was overlooked by some readers. In complicated automatic machines a designer is forced to use cams, instead of other mechanisms, because they give greater freedom in selecting all elements of a desired movement, including the period of forward movement, rest periods, and the period of the return stroke. Quite often the requirements for the return stroke are entirely different than for the forward stroke. Consequently the analysis given in the article was limited to one stroke of the cam.

Each acceleration or speed curve has as a base the time required for one stroke, the cam follower starting from rest and ending at rest. The article describes a method of finding a cam profile for any desirable speed or acceleration curve. Since the "trapeze" cam is not necessarily the most convenient for all application, it is referred to only as one possible solution.

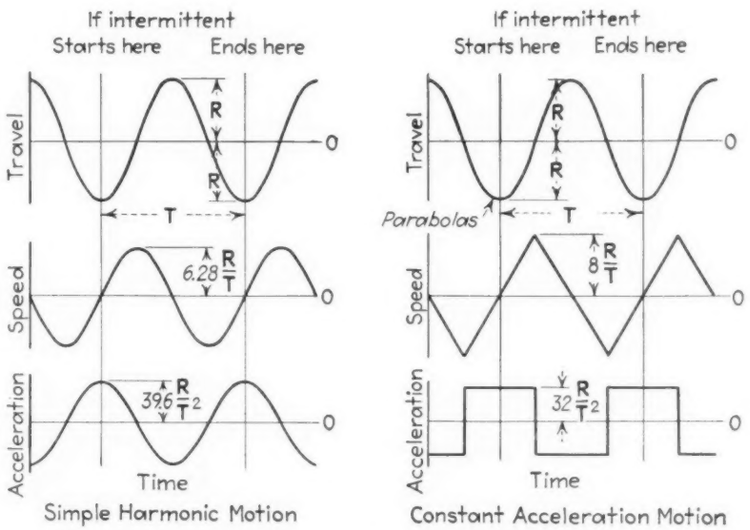
It is quite difficullt to define the word "shock" for all possible cases. Sometimes a shock is the result of an impact when machine parts are worn out, but in some cases a new machine has shocks when its speed is increased, setting up excessive vibration in some part of the machine.

Shock may be produced by the variation of speed, but this statement does not cover all instances. For example, the speed of a body subjected to constant acceleration introduces unvarying stresses and deflections, and although the speed is varying, no shocks are expected. Vibrations can be produced by variation of the applied force, and naturally, the magnitude of the force is important because vibrations cannot be noticed if the deformations and stresses are negligible. However, the method of application of the force should not be overlooked.

When a string supporting a weight is cut, the stresses disappear without any visible effect. But if the weight is suspended by a spring which is hung from a cord, violent vibration will appear in the spring when the cord is cut. A similar effect is found when a long bar is suspended from the center.

Machined parts may vibrate when the applied force is removed faster than the deformations can disappear, so there is another chance for sustaining or increasing vibrations. Variation of acceleration means the variation of inertia force, so the cam imposing sufficiently high acceleration with abrupt variations of acceleration can also produce vibrations. To prevent these sudden changes in acceleration, the "trapeze" cam has a slightly higher maximum acceleration than an equivalent constant-acceleration cam, but the acceleration does not start with the maximum value, nor does the deceleration have a zero value at the end of the stroke. Thus, vibrations will be reduced.

—C. N. NEKLUTIN
Chief Engineer
Universal Match Corporation



Constant acceleration contours give higher velocity and lower acceleration than obtained from simple harmonic motion

Double Dimensioning

To the Editor:

● Ordinary shop practice in dimensioning a shaft detail, for example, would be to furnish two lines of dimensions along the length of the shaft. Dimension of the lengths of the keyways and the intervening blank spaces would total up to the overall length, also given.

These two dimensions give the workman a check on his work. When laying out the keyways and intervening spaces, if the total does not check with the overall length, either the drawing is wrong or an error has been made in laying out the work.

This drafting practice is not tolerated in the automobile industry, where drawings are always in a state of flux. A draftsman may change one detail dimension and not change others affected.

The practice of leaving out the "closing dimension" and locating a keyway by an end dimension is good practice whenever drawings are constantly being changed. The same holds true for a milled opening, for example, straddling a center line. The detail drawing gives only the total width of the opening, the slot being located from the center line. —J. S. CARPENTER
Detroit, Mich.

What is Industrial Design?

To the Editor:

● Industrial design is the design of products for appearance and salability. An authority on marketing says, "Design is fundamentally an engineering problem. It cannot be delegated to persons unacquainted with production machinery and processes. The ideal designer is a type of engineer who can visualize and correlate problems of both consumption and production."

Along with his other qualifications, the industrial designer must have an understanding of the needs and tastes of the consumer. He must know what constitutes good mechanical design; he must have an appreciation of artistic eye appeal, and know production methods and materials fabrication.

Mechanical engineers with artistic ability and knowledge of production methods can act as an intermediary between the industrial artist and the production planners. For only a slight increase in personnel cost, manufacturers could use this type of individual to revamp their designs and step into the field with products which would meet every market requirement for design and sales appeal. —B. S. MALIN
Providence, R. I.

Anti-Friction Bearing

HENRY J. MARTIN

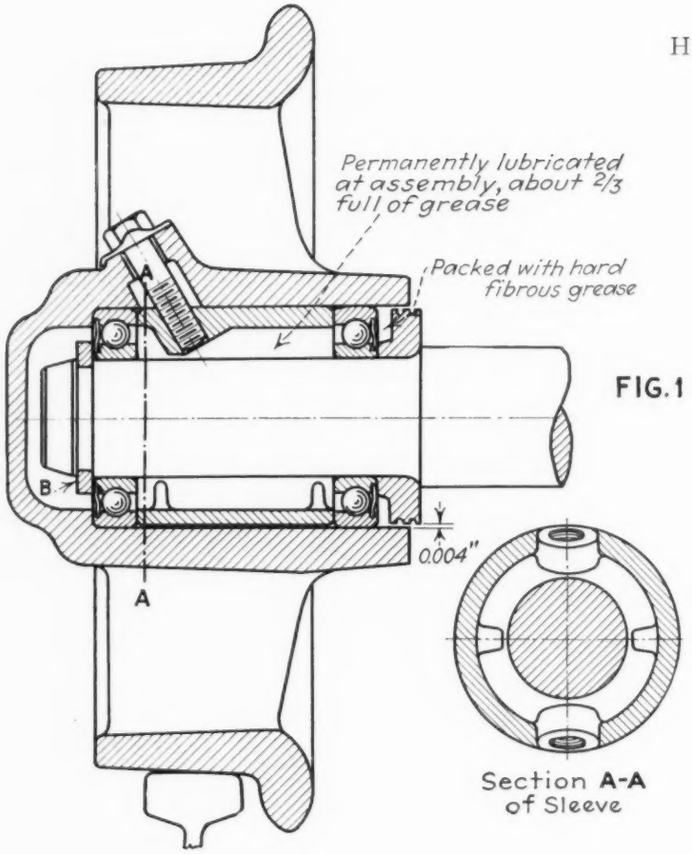


FIG. 1

Fig. 1—Mine car wheel rotates about stationary axle. Bearings and sleeve are held on shaft by snap ring *B*. Wheel hub forms bearing housing and is held on by diagonally placed locked screws which also regulate axial play.

Courtesy Martin-Rockwell Corporation

Fig. 2—Conveyor head and tail pulley is supported in self aligning pillow blocks. Bearings are pressed on split sleeve, ends taper threaded. Nuts clamp bearings and bind sleeve to shaft. Cup and washer grease seals prevent entrance of dirt. Slidable along angle iron way for belt take-up by screw.

Courtesy Timken Roller Bearing Company

Fig. 3—Planetary driving pulley, made of two castings, ball bearing mounted top and bottom in overhanging arm of machine. Three sets of planet gears are mounted in lower half. Lower central pinion, meshing with planets, is supported by one ball bearing. Felt seals protect bearings.

Courtesy New Departure Manufacturing Company

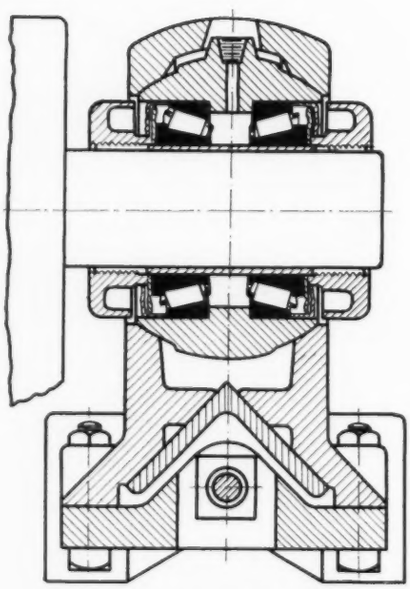
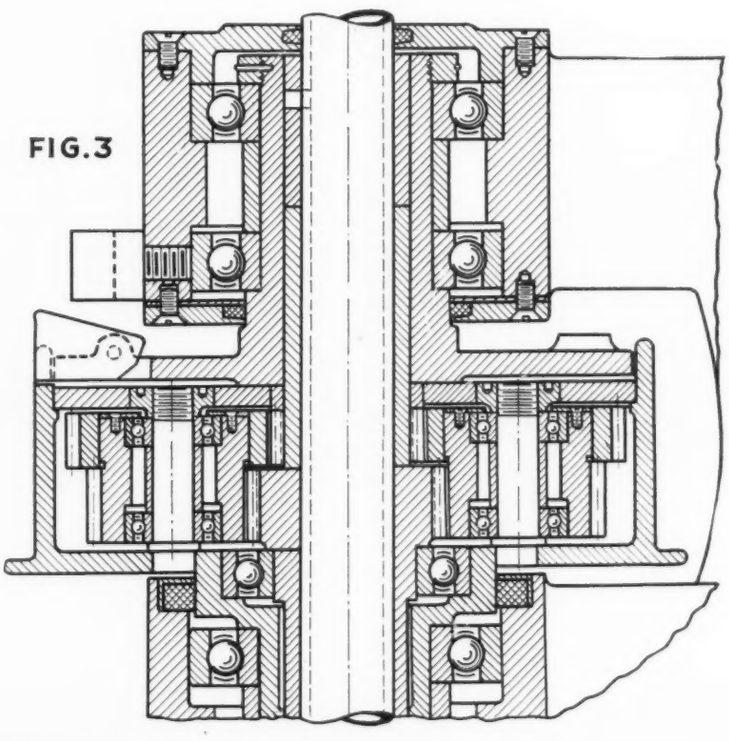


FIG. 2

FIG. 3



Pulley Mountings

Fig. 4—Internal grinding spindle pulley is mounted in sleeves with two Ex-Cell-O double row bearings, and protected by felt seals and retained by double lock nuts. Running at 10,000 r.p.m., bearings are lubricated by sight feed oilers.

Courtesy Ex-Cell-O Aircraft & Tool Corporation

Fig. 5—Multiple vee-belt idler pulley rotates about stationary shaft. Below centerline is shown design for oil lubrication, above for grease. Pulley has 0.005-0.010 in. end play for alignment of running belt. Clamped with one nut.

Courtesy Marlin-Rockwell Corporation

Fig. 6—Mine car wheel rotates about one built-in seal and one plain ball bearing. Combination grease and felt seal in cap retains the lubricant and keeps out foreign material.

Courtesy Marlin-Rockwell Corporation

Fig. 7—Large idler pulley runs on built-in seal bearings pressed on non-rotating shaft. Bearings have thin collar assembled in outer race with small clearance over inner ring, thereby sealing-in lubricant. Pocket between bearings filled with grease when assembled. Pulley assembly is supported on rods *A-A*.

Courtesy Marlin-Rockwell Corporation

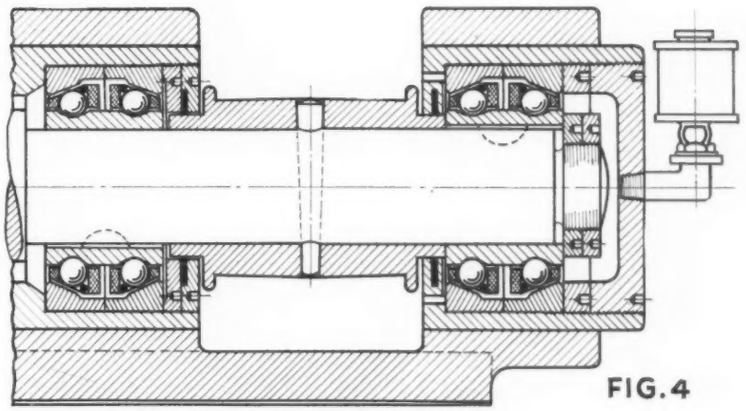


FIG. 4

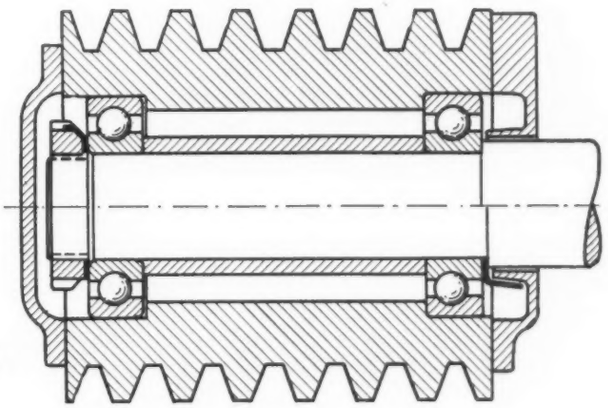


FIG. 5

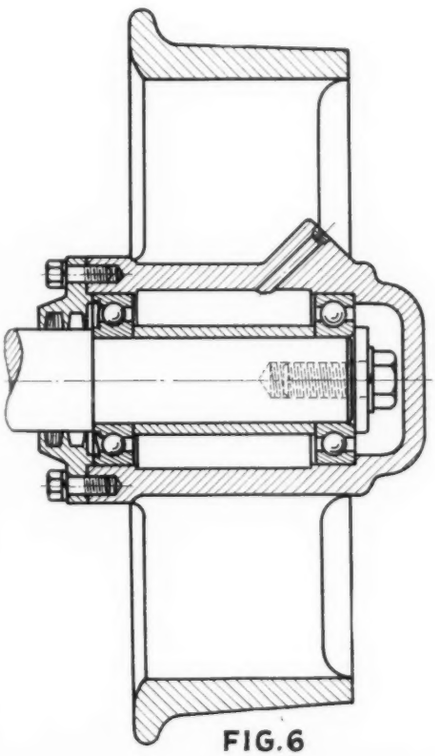


FIG. 6

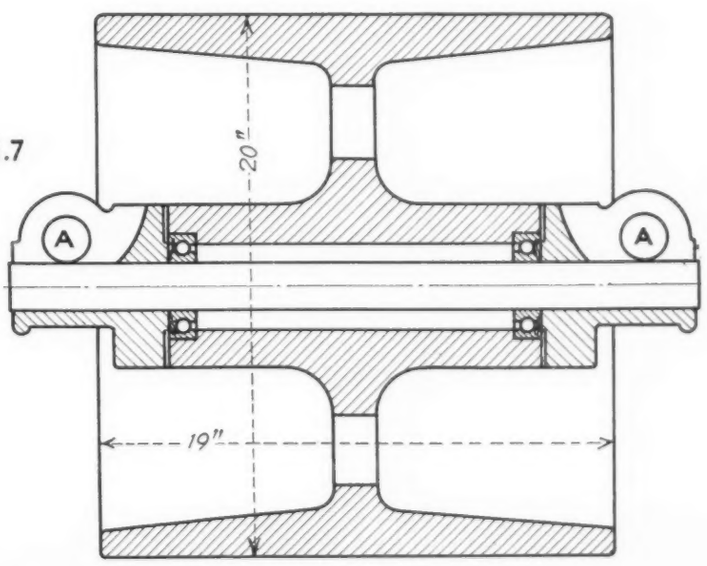
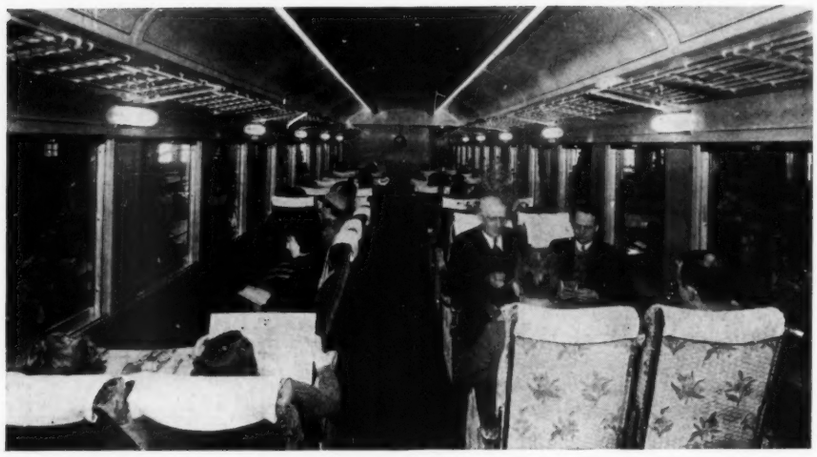


FIG. 7

NEWS



Built by Bethlehem Steel for the Norfolk and Western Railway Company

New Designs of Railway Equipment Are Compelling More Buying

Diesel-electric trains set new speed records. Streamlined electric and steam locomotives, 3,600 hp. diesel electric locomotives, deluxe day coaches obsoleting old equipment

THOSE who contend that revival of the capital goods industries hinges on railroad purchasing should be extremely optimistic over the developments of the past year. Air-conditioned cars, light-weight streamlined unit trains, diesel-electric powered rail cars, unit trains and locomotives, deluxe coaches and electrification have become firmly entrenched.

Diesel electric power units driving light-weight streamlined trains, such as the stainless steel Burlington Zephyr and the Union Pacific aluminum train, point the way to diesel-electric passenger locomotives for main line operation. The Winton Engine Company, using General Electric equipment, now has under construction a two-unit diesel-electric locomotive with a total capacity of 3,600 hp., the largest of this type ever built. An 1,800 hp. locomotive using similar equipment is also under construction for the Baltimore and Ohio Railroad.

At the Erie Works of the General Electric Company is being built for the Illinois Central a 2,000 hp. switching locomotive using a Busch-Sulzer diesel engine to drive the generators. Boston and Maine, Erie and Lackawanna have ordered or have put into service during the past year, similar locomotives.

Satisfaction over the performance of the three-section streamlined aluminum train built by Pullman for Union Pacific is evidenced by the railroad's order for additional units. One of them a six-section train, three of the sections being sleeping cars, was recently put in service. Two nine-section units, 1,200 hp. are under construction. Similarly, Burlington ordered additional stainless steel units from Budd; also an extra section for the Zephyr to take care of the unexpected increase in passengers using the train. Boston and Maine is getting a three-section 600 hp. stainless steel unit from Budd.

Other roads getting diesel electric streamline units include Gulf Mobile and Northern, Illinois Central; a "rail Zeppelin" is being built by Goodyear-Zeppelin for the New Haven.

Pennsylvania's new streamlined 4,620 hp. Westinghouse equipped electric locomotives for operating speeds of 90 m.p.h., and two streamlined steam locomotives being built by The American Locomotive Company for the Chicago, Milwaukee and St. Paul, will set new records for power and speed.

Deluxe day coaches, another 1934 railroad development, promise to obsolete much old equipment. The deluxe coaches built by Pullman-Bradley for the New Haven and styled by Walter Teague were delivered in December. These deluxe coaches are air-conditioned, have sealed windows set in aluminum sash, floors rubber-tiled on cork, and rubber-equipped trucks for better riding and noise reduction.

The Norfolk and Western deluxe coach, built at the Harlan plant of the Bethlehem Steel Company, has rotatable-reclining seats as shown in the picture. Placed on 47 in. centers they allow lots of leg room. Built in pairs, the seats are individually adjustable to reclining positions. Other features include concealed lighting, aluminum baggage racks and men's and ladies' rooms with all the conveniences found in a Pullman.

These are merely a few of the current developments that give promise of a railroad reconstruction program that will make history.

Results of Tests on Carboloy

SOME idea of the life of cemented tungsten carbides may be obtained from recent tests made by Carboloy Company, Inc. In a test to indicate resistance to wear, a bar of Carboloy (91 Rockwell "A") and a bar of hard steel (65 Rockwell "C"), both 0.366 in. square, were sandblasted steadily for 40 min. each. At the conclusion of this test the steel bar was approximately 50 per cent severed while the tungsten carbide bar showed no wear other than a slight dulling of its surface finish.

In another test, bars of drill rod, stainless steel and Carboloy were immersed in concentrated hydrochloric acid for a period of one week. At the end of this period the following loss in content was recorded:

Drill Rod	*100.	per cent
Stainless Steel	47.7	per cent
Carboloy	0.304	per cent

*(consumed completely)

Although the Carboloy bar showed a loss in content of 0.304 per cent, its

surface was not visibly affected. This high degree of resistance to both abrasive and corrosive wear indicates an extensive field for the economical application of Carboloy cemented carbide for this purpose. The following applications, which are among those now in economical use, may suggest a number of further possibilities to the design engineer: These include plug, cylindrical, gagematic and snap gages, indicator points for dial gages, micrometers, etc.; guides of all types such as wire guides, guides for leather and other textile machines; work rests, lathe centers, centerless grinder rests; valve stems and seats, guide block bushings for screw machines, nozzles for abrasive or corrosive materials and fluids.

Trade Fairs to Emphasize International Relations

Initial exposition of International Trade Fair in New York, Feb. 4-9. Leipzig Fair, March 3-10, expects 8,000 exhibits

ONE of the features of the first showing of the International Trade Fair to be held in the Grand Central Palace in New York, Feb. 4-9, will be general composite exhibits instead of the individual booth exhibits commonly found at expositions. This intention is to evaluate a greater appreciation of the particular industry. The development of processes and the history and evolution of certain industries will be dramatized by pageants and manufacturing machinery in operation.

Apparently the major portion of the exhibits will be on clothing, wearing apparel, furnishings and similar non-mechanical products, although it is planned to have a section covering business and household machinery.

The Leipzig Trade Fair to be held March 3-10 will, it is announced, have more than 8,000 exhibits contributed by a score of the leading countries.

At both the International and the Leipzig trade fairs, a special effort will be made this year to extend international trade relations with foreign countries. At New York the public will be shown the need for an increased import business through exhibits and charts demonstrating how America's export business is being damaged by the impossibility of foreign countries to buy if they cannot sell. Similarly at the Leipzig Fair, the results of a study of the requirements of overseas countries is being used to lead the way in encouraging foreign trade.

Homogeneous Steel Ten Times Stronger

IN his paper "Steel and Cast Iron in Stress-Transmitting Mechanisms," presented at the last annual meeting of the American Society of Mechanical Engineers, Everett Chapman, vice-president of the Lukenweld division of the Lukens Steel Company, included a highly interesting example of the possible great increase in the strength of a material by attaining complete homogenity.

As explained by Mr. Chapman, the cohesive forces between the atoms making up a material is a measure of its theoretical strength. This atomic bond can be calculated from a laboratory measurement of the latent heat of vaporization of a given material and, assuming that the same degree of cohesion exists at room temperatures as exists at the vaporizing point, will give a value for the atomic bond of at least the right order of magnitude. Griffith, in England, measured these values for common glass. The measured strength of the glass chosen was 25,000 lb. per sq.in. This is the average strength of the gross mass with all its internal discontinuities and internal strains as present in a tensile test bar. The atomic forces, as measured by the latent heat of vaporization, demanded that the material show a strength of around 500,000 lb. per sq.in.

Griffith then broke fine threads of this material, measuring the stress at rupture. Fragments of the same thread were then broken and measured, and the fragments resulting from this test were again broken and measured. His final test specimen, selected on the basis of tensile elimination of internal flaws and defects that tended to raise the average stress, broke at 491,000 lb. per sq.in. This is a remarkably interesting indication of the possible strength of materials in the homogeneous state.

The atomic bond in pure iron, as theoretically deduced from its latent heat of vaporization, indicates that the strength of steel should be about ten times the observed values in ordinary commercial steel. The difference between the observed values and the theoretical values can be reasonably supposed to be due to stress-raising, non-metallic impurities, minute internal cracks resulting from cooling, and internal strains distributed at random that pre-load the material from point to point.

A.I.E.E. to Discuss Electronic Devices

ONE of the two sessions on electronic devices at the annual meeting of the American Institute of Electrical Engineers, being held Jan. 22-25, is to be given entirely to a discussion of industrial applications. Of particular interest are two papers, one by F. H. Gulliksen and R. N. Stoddard on "Industrial Electronic Control Applications," and another by D. E. Chambers on "Industrial Applications of Electron Tubes."

MEETINGS

Society of Automotive Engineers—Annual meeting, Book-Cadillac Hotel, Detroit, Mich., Jan. 14-18. John A. C. Warner, secretary, 29 West 39th St., New York, N. Y.

American Institute of Electrical Engineers—Annual meeting, Engineering Societies Bldg., New York, N. Y., Jan. 22-25. H. H. Henline, secretary, 33 West 39th St., New York, N. Y.

American Institute of Mining and Metallurgical Engineers — Institute of Metals Division, Engineering Societies Bldg., New York, N. Y., Feb. 18-21. A. B. Parsons, secretary, 29 West 39th St., New York, N. Y.

EXHIBITIONS

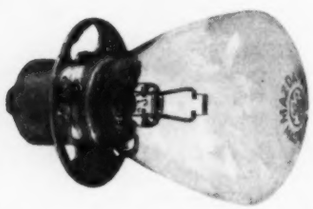
National Automobile Show — Grand Central Palace, New York, N Y., Jan. 5-12. Alfred Reeves, manager, 366 Madison Ave., New York, N. Y.

International Trade Fair—Grand Central Palace, New York, N. Y., Feb. 4-9. International Trade Fair Inc., 480 Lexington Ave., New York, N. Y.

New Materials and Parts

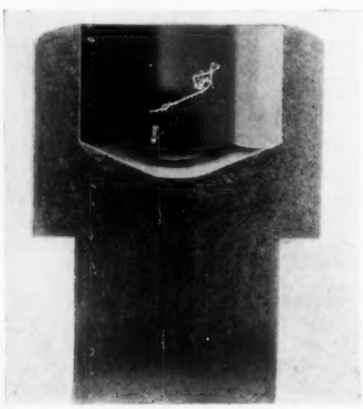
Single Filament Lamp With Prefocussed Base

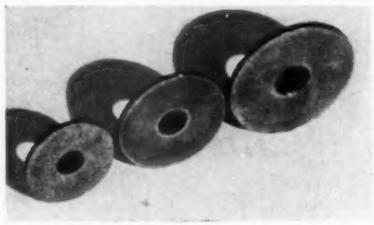
Prefocussed lamp with a single filament designed for spot lighting, for local lighting on machines or for photoelectric controls. For use only in specially designed socket in which the collar seats against a flange on the socket. Filament is centered and aligned with the collar in manufacture by soldering the collar to the base after adjusting the filament position. Collar has three bayonet slots engaging pins in the socket. Design is said to insure more accurate light control and to give accurate filament alignment. General Electric Co., Nela Park, Cleveland, Ohio.



Socket Screw

Forged to obtain continuous fibers from the wrench socket into the screw body, it is claimed that "Fibro Forged" screws have increased strength in all portions of the screw where most needed. Photograph of etched sample shows fibers running up cup-shaped from the screw body and terminating in walls of socket. Holo-Krome Screw Corp., Bristol, Conn.

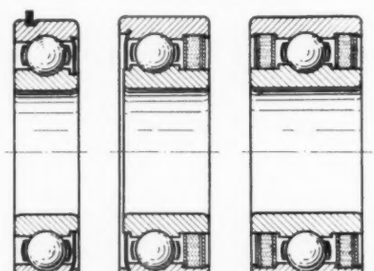




Aviation Washers

Large outside diameters to conform to U. S. Army and Navy aviation standards are made of wrought iron and steel, stainless or tinned steel, brass, copper, aluminum, and aluminum alloys. Small pattern washers of same material also available. Wrought Washer Mfg. Co., 2100 S. Bay St., Milwaukee, Wis.

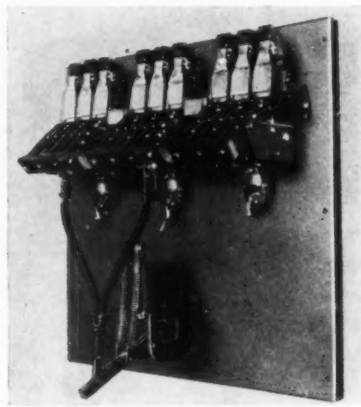
Ball Bearings



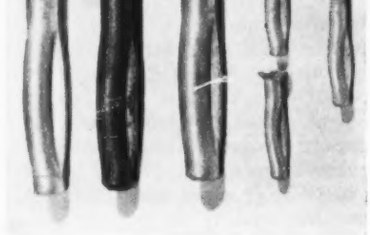
To eliminate a shoulder in the bearing housing, a steel snap ring is inserted in a groove in the outer ring, close to one face. Said to reduce machining cost and give a more compact mounting. One side is shielded to retain grease. Also available without the shield. In another series, removable felt seals between metal plates, for grease retention, are assembled into the outer race. Furnished either with one felt seal, one felt seal and one metal shield, or with two felt seals. Norma-Hoffmann Bearings Corp., Stamford, Conn.

Air Circuit Breaker

Is of the one contact per pole type. A stationary, laminated brush mounted on the panel is contacted by a moving arm, completing the circuit. Design is said to reduce resistance and heating losses. Available in capacities up to 10,000 amp., 750 volts, a.c., and 600 volts d.c. Interrupting capacity is from 40,000 to 120,000 amp. Said to carry its rated current continuously with a temperature rise of not more than 20 deg. C. on the contacts. Roller-Smith Co., 233 Broadway, New York, N. Y.



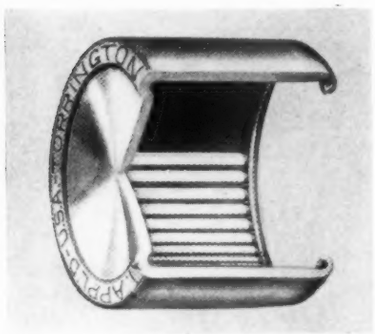
Expansion Bolt



A holding or anchoring device for attaching machines to concrete floors, brick, stone or other masonry. The bolt body is sheared longitudinally through the center and bowed outward on both sides. Heat-treatment is said to give the sheared portion spring properties so that tension secures the grip. It is driven with a hammer. Round and countersunk type heads available for permanent holding and stud type where removal of machine is necessary without disturbing the bolt. Various finishes are obtainable. The Rawlplug Co., Inc., 98 Lafayette St., New York, N. Y.

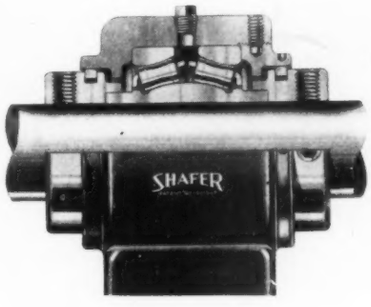
Needle Bushing

With closed end and made of one piece outer shell roller raceway which is cyanide hardened steel. Other end has turned-in lip to retain the rollers and to afford channel for sufficient lubrication. Used as end bearing for shaft and designed to retain oil and grease. Made in a complete unit it is said to facilitate assembly and disassembly. The Torrington Co., Torrington, Conn.



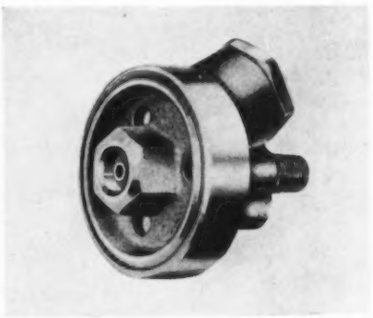
Piston Ring Sealed Bearing

The inner race of this roller bearing has four flats on its outside diameter at either end which engage two locking collars clamped to the shaft by two set screws. Each collar carries a piston ring which is said to provide an effective seal against dirt, dust and water. The piston ring seals are furnished on double row radial-thrust self-aligning roller bearings. Shafer Bearing Corp., 6501-99 West Grand Ave., Chicago, Ill.



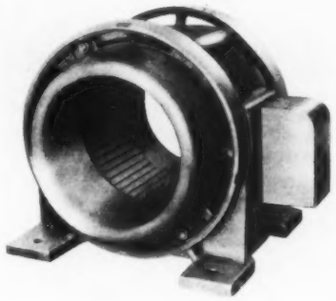
Self-Cleaning Nozzle

Used on water supply for air conditioning and humidifying. Nozzle body made of brass. A stainless steel diaphragm supports a movable orifice through which a fixed needle projects when water supply is closed. When supply is opened, the water forces diaphragm outward and moves orifice away from needle. Nozzle furnished with $\frac{1}{8}$ in. pipe thread with pressure of 30 to 80 lb. and 1.8 to 2.9 gal. per hr. Detroit Lubricator Co., Detroit, Mich.



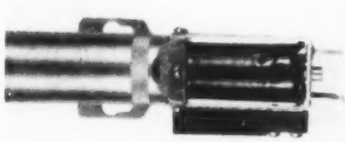
Seal-Clad Motor

Squirrel-cage, open type construction with permanent coil protection. Bakelite shields are sealed over the stator coil to give protection against metallic dust, grit, oil, moisture and mild acids injurious to insulation. Frames are made of cast steel; rotors are silver-brazed. Available with oil and dust-tight sleeve, or anti-friction bearings, in capacities up to 25 hp., 1800 r.p.m. Allis-Chalmers Mfg. Co., Milwaukee, Wis.



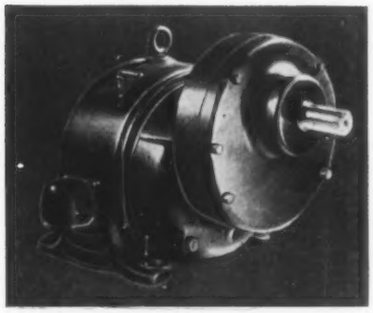
Delayed Action Relay

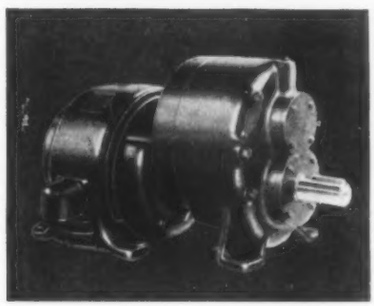
Is of the oil-dashpot type, operating on alternating current. Requires no series condenser. The time delay interval between the closing of the coil circuit and the operation of the contacts can be varied from 5 sec. to $1\frac{1}{2}$ min., by turning the outer dashpot cylinder. The non-adjustable release time is said to be 0.010 sec. It is furnished to operate on various voltages and frequencies and can be provided with a variety of contact combinations. American Automatic Electric Sales Co., 1033 W. Van Buren St., Chicago, Ill.



Geared-Head Motors

Single parallel geared motors ranging from 6:1 reduction to 1:2 speed increase and rating from $\frac{1}{10}$ to 100 hp. are available. Double parallel geared motors ranging from 8:1 to 36:1 reductions, and ratings from $\frac{1}{10}$ to 75 hp. are available. Herringbone gears are used except in smaller sizes. They





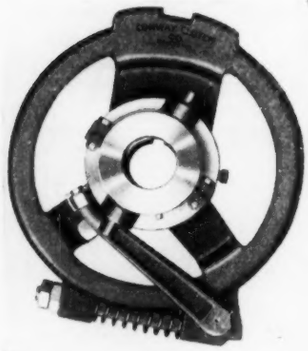
are mounted on splined shafts in the single type. Gears and shafts are made of heat-treated alloy steel. Motor shaft runs on ball bearings and gear shafts are mounted on tapered roller bearings. Patented oil seals are used on all shafts. Geared head may be mounted in any of four positions around motor shaft in the single type. In the double type, motor shaft and final drive shaft are on the same centerline. Special mountings for vertical drives are obtainable. The Master Electric Co., Dayton, Ohio.

Fan-Cooled Motors

Totally inclosed d.c. motors designed for applications where dust, dirt, moisture or other foreign matter is present in large quantities. Motors are totally inclosed and fan-cooled, available in sizes from $\frac{1}{2}$ to 200 hp. Watertight conduit boxes, labyrinth seals and a system of dual ventilation are

used. An internal fan draws warm air from the windings and core, circulating it around the frame and end shield. An external fan, drawing air through mesh openings in end shield, cools the magnet frame and inner end shield. Motors are said to be interchangeable with most open motors of corresponding rating. General Electric Co., Schenectady, N. Y.

Conway Clutch



Compact compression type clutch for hoists, winches, drums, brakes and cables. Has floating, asbestos friction band, actuated by a double-action leverage mechanism. Hard roller on lever contacts cone. Friction band, latch and lever can be removed readily for replacement. Extension sleeve available. The Conway Clutch Co., 1543 Queen City Ave., Cincinnati, Ohio.

Dial Indicators

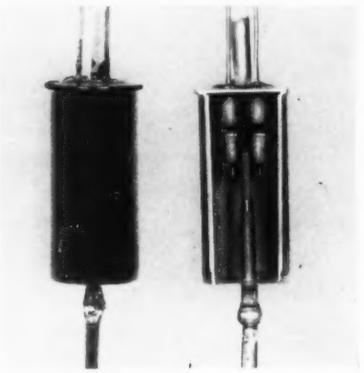
Readings either thousandths, half-thousandths, ten-thousandths inches, or hundredths and five-hundredths of a millimeter. Working parts are made of stainless steel; the case and bezel are chromium plated. The back is a solid die casting recessed to hold the lug for clamping. The crystal is non-breakable. Sapphire jewels are used. Dials are marked with black figures against a white background. The L. S. Starrett Co., Athol, Mass.

Adjustable Transformer

For reducing voltage for devices requiring considerable power. Of the variable-voltage type, transformer is supplied in an open iron frame. Instrument will control at any setting, a load which draws up to 2 kva. at line voltage. Available for 115 and 230 volts supply, 50 to 60 cycles. General Radio Co., 30 State St., Cambridge, Mass.

Metal Vacuum Switch

Made almost entirely of steel and designed to take advantage of the absence of an arc when breaking a circuit in a high vacuum. Switch is about ¾ in. in diameter, 1½ in. long, and can be operated by a fraction of an ounce of pressure. Switch casing is made of steel, hydrogen-copper-brazed and exhausted to a high vacuum through a hollow steel operating stem which passes into the contact chamber through a thin steel end-wall. Lead-in wires to the contacts pass through glass beads sealed into the chamber with a gas-tight joint. Switch rated at 10 amp., 250 volts d.c. or 440 volts a.c.; or 5 amp., 500 volts d.c. Said to be capable of interrupting as much as 5 hp. as fast as 30 times a sec. General Electric Co., Schenectady, N. Y.

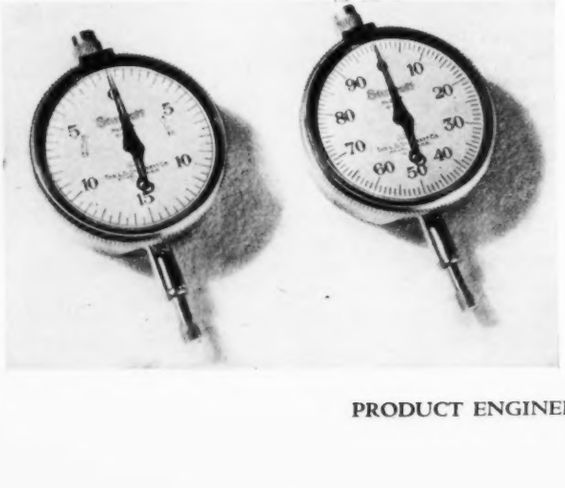


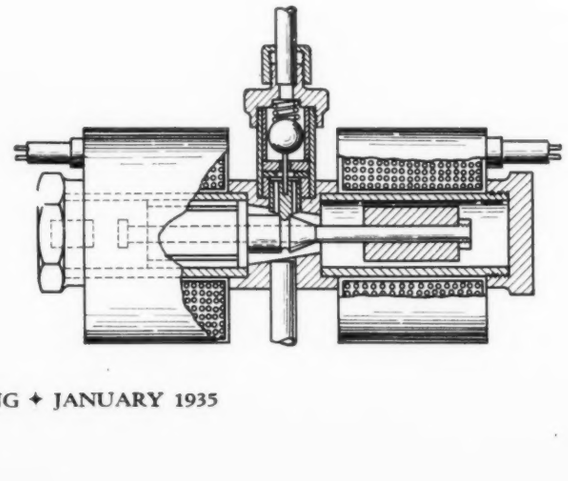
Quick-Acting Relay

Operating on a.c., has a shading ring said to provide chatterless contact closure. Relay furnished for operation on any voltage from 6 to 220 volts, 50-60 cycles. Available with a variety of spring combinations. Operation and release periods approximately the same as for corresponding d.c. relays with similar spring loads. Slip-on cover of sheet steel, aluminum finished, can be supplied. American Automatic Electric Sales Co., 1033 W. Van Buren St., Chicago, Ill.

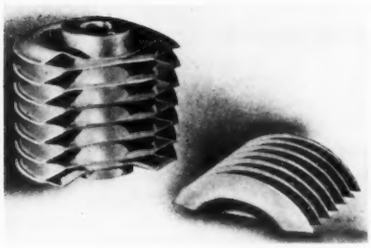
High-Pressure Valve

Solenoid operated packless valve designed for use with liquids at high pressure such as liquid carbon dioxide. Valve is of the momentary type and is designed to operate under pressures up to 1,500 lb. per sq.in. Hoppe Engineering Co., 246 So. Meridian St., Indianapolis, Ind.





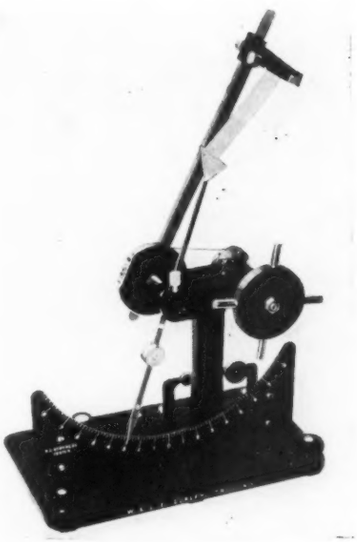
V-Belt Sheave



Made of steel with the outside walls strengthened by convex plates to withstand distortion. The rim, web and interior grid-type construction are welded for rigidity. Allis-Chalmers Mfg. Co., Texrope Div., Milwaukee, Wis.

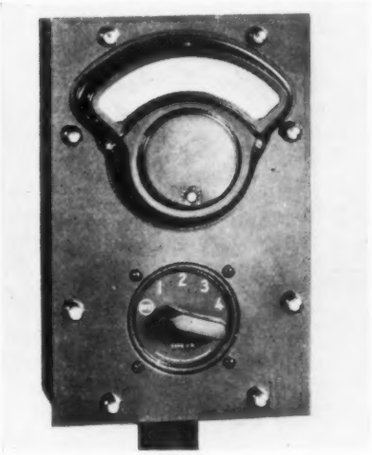
Paper Stiffness Tester

Measures the stiffness, softness, and pliability of all kinds of paper. Test pieces clamped in the movable arm are guided over the top of the weighted pointer, until the bending of the paper releases the pointer. Deflection is expressed in milligrams on the scale at the bottom of the instrument. W. & L. E. Gurley, Troy, N. Y.



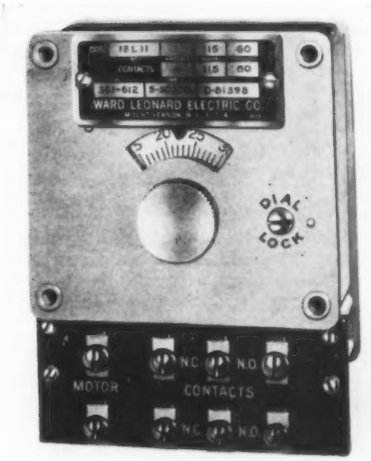
Low Resistance Pyrometer

Used for recording temperatures of small diesel engines and gas engines in two ranges, 80 deg. to 1200 deg. F., on a scale 3¼ in. long. Housed in a pressed steel case on a bakelite front panel. Said to be moisture tight. Furnished in two styles for indicating temperatures from several sources at a central point, or from one source only. Also furnished without steel box or panel, attached to cast aluminum bracket. Thwing Instrument Co., 3317 Lancaster Ave., Philadelphia, Pa.



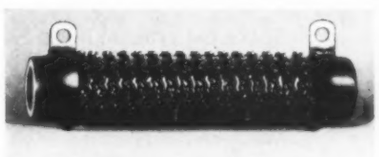
Time Delay Relay

Incorporates a synchronous motor running in oil, a clutch with low-voltage release magnet, and a dial lock. Silver-to-silver contacts are provided in all single and double pole and double pole double throw combinations. Motor casing is made of a thin brass shell and cap soldered together after assembly. Oil hole in rotor neck is soldered shut after lubricating oil has been injected, insuring that all rotating parts of motor are totally inclosed. Dial lock prevents any unintentional change in time interval setting. Ward Leonard Electric Co., Mt. Vernon, N. Y.



Corrugated Ribbon Resistor

Is made of vitreous enameled units in which a corrugated ribbon of resistance metal is spacewound edgewise around a porcelain tube. Ends of the ribbon are locked and brazed to the terminal lugs or ferrules. Vitreous enamel fills the space between the



turns of the ribbon and holds each in place. For high wattage applications, with resistance up to 10 ohms. Ohmite Mfg. Co., 636 N. Albany Ave., Chicago, Ill.

MANUFACTURERS' PUBLICATIONS

Alcumite—The Duriron Co. Inc., Dayton, Ohio. Bulletin 701, describing chemical corrosive resistance of copper base aluminum-iron alloy and showing pictures and graphs.

Ball Bearing—The New Departure Mfg. Co., Bristol, Conn., Booklet O, 14 pages, 3½x6¼ in. "Let's Can the Oil Can," describes characterstics of permanently sealed bearings. Application and mounting data also given.

Cast Dies—Meehanite Metal Corp., Pittsburgh, Pa. Bulletin 3 shows use of cast metal for punches and dies. Illustrated with applications and stamping examples.

Circuit Breakers—Roller-Smith Co., 233 Broadway, New York, N. Y. Catalog 5, 14 pages "Air Circuit Breakers," describes the mechanism of a heavy duty type.

Enameling Iron — The American Rolling Mill Co., Middletown, Ohio. Pamphlet "Facts You Should Know About Armco Enameling Iron," describing characteristics and listing uses.

Fiber—Spaulding Fibre Co., Inc., Tonawanda, N. Y. Pamphlet "An Improved Art for Radio" describing and illustrating tubing for coil forms.

Insulation Testing Equipment — Sound Engineering Corp., 416 N. Leavitt St., Chicago, Ill. Bulletin 934, 12 pages, 8½x11 in. describing dielectric testing instrument and auxiliary equipment.

Microscopes—Bausch & Lomb Optical Co., Rochester, N. Y. Booklet,

"Polarizing Microscopes," 6½x9¼ in., 25 pages, explaining application of microscopy to various materials. Illustrated.

Motors—Master Electric Co., Dayton, Ohio. Data 3710, "Uni-Brake," Data 4525, "Single Parallel Geared-Head Motors" and data 5025, "Double Parallel Geared-Head Motors," 8½x11 in. describing multiple-disk type of motor brake and including engineering data. Herringbone gear motor reducers in ratios up to 36 to 1 explained. Photographs and tables included.

Power Feed—Oilgear Co., 1301-1417 West Bruce St., Milwaukee, Wis. Bulletin 44100 "New Fluid Power Feeds," 19 pages, 8½x11 in. describing pumps for adjustable feeding speeds on machines and for mechanisms employing repetitive motion sequences hydraulically. Illustrated with application photographs and schematic diagrams.

Pyrometers — Thwing Instrument Co., 3317 Lancaster Ave., Philadelphia, Pa. Leaflet A-3410. Illustrates and describes a small-sized, low-resistance pyrometer for use in diesel and gas engine work.

Seals—National Motor Bearing Co., Inc., 1100 78th Ave., Oakland, Calif. Catalog, 16 pages, showing industrial applications of oil, fluid and grease seals, featuring new type of leather seal. Also special expansion seals.

Steel—Bliss and Laughlin, Inc., Buffalo, N. Y. Folder, "Ultra-Cut" 4 pages explaining free-machining steel with photographs, micrographs and physical properties.

Steel—Lukens Steel Co., Coatesville, Pa. Pamphlet, 7 pages, 5x7½ in. "Specifications for Chrome-Manganese-Silicon Alloy Steel Boiler Pipe," with chemical and physical properties, various tests, inspection and rejection data.

Steel — Union Drawn Steel Co., Massillon, Ohio. Pamphlet "A Valuable Steel Service" describing special cold working processes of Bessemer screw stock.

Steel Heat-Treatment — E. I. du Pont de Nemours & Co., Inc., Wilmington, Del. Manual, 28 pages, 8½x11 in. Covers procedure followed for case-hardening, re-heating, nitriding and mottling steel with sodium cyanide and describes coloring, tempering and drawing of steels with heat-treating salts. Illustrated with microphotographs.

Synthetic Resin Finishes — Bakelite Corp., Bound Brook, N. J. Booklet, "Integrity of Finishes," 16 pages 6x9 in. Protective coatings for exposure to abrasion, moisture, alkali and salt water. Illustrated.

Tubing—Steel and Tubes, Inc., Cleveland, Ohio. Folder giving mechanical applications of "Electrunite" electric welded tubing and photographs of ductility and strength tests.

BOOKS AND BULLETINS

A.S.T.M. Tentative Standards

1,250 pages, 6x9 in. Paper cover. Published by the American Society for Testing Materials, 260 S. Broad St., Philadelphia, Pa. Price $7.00.

This volume contains all the A.S.T.M. tentative specifications, methods of tests, and definitions of terms covering engineering materials. Of the 236 tentative standards, 48 of these are included for the first time, while 60 of these were revised during 1934. New proposed standards cover alloy-steel castings for high temperature, alloy-steel pipe, copper-silicon alloy rods, sheets, bars and shapes, and magnesium-base alloy die castings. To facilitate the use of the book a complete subject index has been included and two different tables of contents are given.

Logarithms

N. R. Corke. 80 pages, 8½x11 in. Blue clothboard covers. Published by Gee & Co., Ltd., 6 Kirby St., London, E.C. 1, Price 9/3.

Written to teach calculation by logarithms, the text covers the use of tables and slide rules, plotting of curves on semi-log and log graph paper and the construction of alignment charts. The book is designed for self instruction, and will be of practical interest to designers who are taking up the study of logarithms. Four-place tables are included in an appendix.

Machinery's Handbook Ninth Edition

Erik Oberg and Franklin D. Jones. 1,592 pages, 4½x7 in. Published by The Industrial Press, 148 Lafayette St., New York, N. Y. Price $6.

In the ninth edition of this handbook there are 172 revisions, which include new sections and changes in data made necessary by improved knowledge or technique. Major revisions consist of new strength of material data; new data on load capacities of ball and roller bearings; recent developments in plain bearing design; new tables and information on design of various classes of gearing, including spiral bevel and hypoid gears; revised SAE standards; additional screw thread data; a new section on roller chain transmission; revised speed and feed data including actual practice on cemented-carbide tools; new standards for taps, gages, and milling cutters; hardness conversion table; electric motor characteristics of value to machine designers; die-casting compositions; and others. The handbook is divided into 126 main subjects, and the items in these divisions can be located by referring to a complete index. A thumb index is of assistance in opening the book to any one of fourteen sections.

Lubricating Oil Tests

Virginia Polytechnic Institute, Blacksburg, Va., *Bulletin 33* by James I. Clower.

A 41-page booklet showing the significance of chemical, physical and mechanical tests. In the first group, tests were made for acidity, carbon residue, saponification, and sulphur. Physical tests were made for cloud, color, corrosion, emulsification, flash, precipitation, gravity and viscosity. Mechanical tests were made for extreme pressure, wear and lubricant value. SAE numbers and Navy symbols are explained. Saybolt universal viscosity table is included.

Product Development

Metropolitan Life Insurance Co., New York, N. Y. Policyholders Service Bureau. *Bulletin 414*, 23 pages.

Reporting how 75 companies, representing a cross-section of industry, have organized their search for new products, and their procedure for perfecting them in accordance with preferences of consumer.

The survey reveals that these companies in the selection of ideas for development, give attention to such considerations as: (1) whether the product is suitable for marketing to establish sales outlets; (2) whether it can be fabricated with existing plant and equipment; (3) whether it will overcome seasonal dullness; (4) whether the potential market is worth cultivating.

The survey emphasizes the need of organization and systematic procedure, and that product development is not only concerned with mechanical excellence but also with qualities in the product and market outlet.

The Bearing Value of Rollers

University of Illinois, *Engineering Experiment Station, Bulletin No. 263*, by W. M. Wilson, 32 pages, 40 cents.

As the basis for design formulas, the bearing value of a roller can be given only in arbitrary terms. Results of static and rolling tests on rollers from 2 to 120 in., in diameter, made of several types of steel with various heat-treatments, are given in tables and charts.

Gear Design Calculations–I

Spur Gears

CHARLES S. DE LATE

In these examples it will be assumed that the load is applied at the tip of the tooth, and that the number of teeth, diametral pitch and pressure angle are given.

Problem—An engine develops a torque at 270 ft.-lb. at 1,200 r.p.m. and 200 ft.-lb. at 3,200 r.p.m. The clutch gear has 25 teeth and the constant mesh gear 39 teeth, of 8 D.P. and 20 deg. stub tooth form. Compute:

A—The face width under both speed conditions to give a working fiber stress of 60,000 lb. per sq.in. by the Marks and Cutter equation.

B—The maximum static load for wear and the dynamic load at both 1,200 and 3,200 r.p.m., for measured tooth error of 0.001 in. and hardened gears with Sf = 220,000 lb. per sq.in. limiting compressive stress.

C—The flexural stress if the dynamic load and the transmitted load were applied along the line of action at the outer end of the tooth profile.

A—Calculation of face width:

$$W = \frac{s}{k} p T \left(0.278 - \frac{2.69}{n}\right) v a$$

p = Circular pitch
n = Number of teeth in pinion = 25
v = Velocity coefficient
a = Arc of action coefficient
T = Tooth width
W = Tooth load
s/k = Working fiber stress = 60,000 lb. per sq.in.
s = Modulus of rupture
k = Factor of safety

Values for v and a are taken from empirical data set up for use in Marks and Cutter formula.

Pitch line velocity = Pitch line circumference × r.p.m.

$$= \frac{25 \text{ teeth}}{8\ D.P.} \times \frac{\pi}{12} \times 1{,}200 = 981 \text{ ft. per min.}$$

$v = 0.553 \qquad a = 1.24$

$$W = \frac{\text{Torque in lb.-in.}}{\text{Pitch radius}} = \frac{270 \text{ ft.-lb.} \times 12}{25/(8 \times 2)} = 2{,}075 \text{ lb.}$$

$$2{,}075 = 60{,}000 \times \frac{\pi}{8} \times T\left(0.278 - \frac{2.69}{25}\right) 0.553 \times 1.24$$

T = 0.754 in. tooth width for 270 ft.-lb. at 1,200 r.p.m.

For 200 ft.-lb. torque at 3,200 r.p.m.,

$$\text{Pitch line velocity} = \frac{25}{8} \times \frac{\pi}{12} \times 3{,}200$$

$$= 2{,}620 \text{ ft. per min.}$$

$v = 0.39 \qquad a = 1.24$

$$W = \frac{200 \times 12}{\frac{25}{8 \times 2}} = 1{,}532 \text{ lb.}$$

$$1{,}532 = 60{,}000 \times \frac{\pi}{8} T\left(0.278 - \frac{2.69}{25}\right) 0.39 \times 1.24$$

T = 0.790 in. tooth width for 200 ft.-lb. at 3,200 r.p.m.

B—Computation of maximum static load for wear, and the dynamic load:

$W_d = T(W + I)$ = total dynamic tooth load in lb.
T = Tooth width
W = Applied tangential load in lb. per in. of tooth width
I = Increment load in lb. per in. of tooth width

$$I = \frac{0.05 V f}{0.05 V + \sqrt{f}}$$

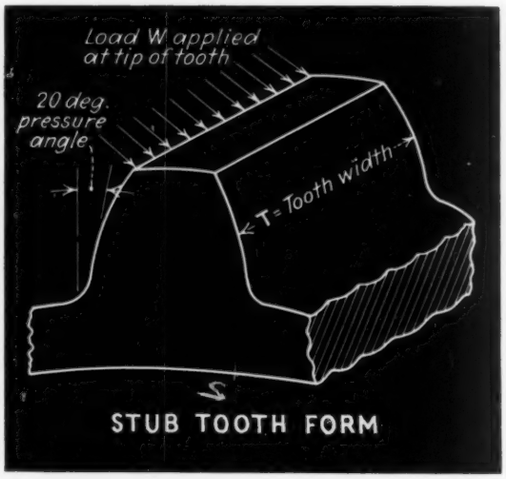
V = Pitch line velocity in ft. per min.
f = Load in lb. per in. of tooth width required to deflect the teeth a distance equal to the tooth error

$$f = W\left(\frac{e}{a} + 1\right) \qquad (1)$$

e = Measured error on gear tooth in inches
a = Tooth deformation on a pair of tooth profiles at the middle of the tooth form

$$a = W\frac{(E_1 Z_1 + E_2 Z_2)}{(E_1 Z_1 \times E_2 Z_2)} \qquad (2)$$

E_1 = Modulus of elasticity for pinion
E_2 = Modulus of elasticity for gear
Z_1 = Elastic form factor for pinion tooth



Z_1 = Elastic form factor for gear tooth
Z may be obtained from a form factor table in a gear book.
$Z_1 = 0.11026$ $Z_2 = 0.11218$

By combining Equations (1) and (2)

$$f = \frac{eE_1Z_1Z_2}{Z_1 + Z_2} + W$$

For 270 ft.-lb. at 1,200 r.p.m.

$$f = \frac{0.001 \times 30{,}000{,}000 \times 0.11026 \times 0.11218}{0.22244} + 2{,}075/0.790 = 4{,}300 \text{ lb. per in.}$$

$$I = \frac{0.05\,V\,f}{0.05\,V + \sqrt{f}} = \frac{0.05 \times 981 \times 4{,}300}{0.05 \times 981 + \sqrt{4{,}300}} = 1{,}840 \text{ lb. per in.}$$

$$W_d = T(W + I) = 0.790\left(\frac{2{,}075}{0.790} + 1{,}840\right) = 3{,}525 \text{ lb.}$$

For 200 ft.-lb. at 3,200 r.p.m.

$$f = \frac{0.001 \times 30{,}000{,}000 \times 0.11026 \times 0.11218}{0.22244} + 1{,}532/0.790 = 3{,}614 \text{ lb. per in.}$$

$$I = \frac{0.05 \times 2{,}620 \times 3{,}614}{0.05 \times 2{,}620 + \sqrt{3{,}614}} = 2{,}480 \text{ lb. per in.}$$

$$W_d = 0.790\left(\frac{1{,}532}{0.790} + 2{,}480\right) = 3{,}490 \text{ lb.}$$

W_s = maximum static tooth load for wear

$$W_s = \frac{2N}{n+N} T d \frac{Sf^2 \sin\phi}{1.4}\left(\frac{1}{E_1} + \frac{1}{E_2}\right)$$

$$\frac{2N}{n+N} = \text{ratio factor} = \frac{2 \times 39}{25 + 39} = \frac{39}{32}$$

T = tooth width = 0.790 in.

d = pitch diameter of pinion $= \frac{25}{8} = 3.125$ in.

$$\frac{Sf^2 \sin\phi}{1.4}\left(\frac{1}{E_1} + \frac{1}{E_2}\right) = \text{stress factor}$$

ϕ = Pressure angle of tooth system
Sf^2 = Limiting compressive stress

$$W_s = \frac{39}{32} \times 0.790 \times 3.125 \frac{220{,}000^2 \times 0.34202 \times 2}{1.4 \times 30{,}000{,}000}$$

$$W_s = 2{,}380 \text{ lb.}$$

C—Compute the flexural stress with load applied at end of tooth

$$W_s = Sp\,Ty$$

W_s = W_d for dynamic loading
W_t = W for transmitted load
p = Circular pitch, inches $= \frac{\pi}{8}$
y = Tooth-form factor. Use 0.133
S = Flexural stress

For 270 ft.-lb. at 1,200 r.p.m.

$$S = \frac{W_s}{pTy} = \frac{3{,}525}{\frac{\pi}{8} \times 0.790 \times 0.133} = 84{,}600 \text{ lb. per sq.in. for dynamic loading}$$

$$S = \frac{2{,}075}{\frac{\pi}{8} \times 0.790 \times 0.133} = 49{,}800 \text{ lb. per sq.in. for transmitted load}$$

For 200 ft.-lb. at 3,200 r.p.m.

$$S = \frac{3{,}490}{\frac{\pi}{8} \times 0.790 \times 0.133} = 84{,}000 \text{ lb. per sq.in. for dynamic loading}$$

$$S = \frac{1{,}532}{\frac{\pi}{8} \times 0.790 \times 0.133} = 36{,}800 \text{ lb. per sq.in. for transmitted load}$$

In the March number will be given examples of bevel gear tooth design with the method of computing pitch diameters, radial and thrust loads, and a detailed procedure for checking the possibility of tooth interference in gear and pinion.